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Second Quarterly Technical Progress Report
Continuation of 3-KW Stirling Cycle Solar Power System Program
(1 August through 31 October 1961)

Contract No. AF33(616)-8332
Project No. 3145, Task No. 30500

Engineering Department Report No. 2381

Engineering Dept., Aircraft Engine Operations
ALLISON DIVISION OF GENERAL MOTORS CORPORATION
Indianapolis, Indiana

TRANSMITTAL NOTICE

Allison Division
of
General Motors Corporation
Indianapolis 6, Indiana

Date November 22, 1961

To: Mr. George W. Brady
Research and Engineering Support Division
Institute for Defense Analysis
Suite 100

Attention: 1825 Connecticut Avenue, N.W.
Washington 9, D.C.

Reference:

ENGINE MODEL ______________

QUANTITY

Specification No. ________ ________ Installation Drawing ________

Brochure, Report ________ ________ Detail Drawing ________

Revision ________ ________ Revision ________

Date ________ Date ________

Remarks:

4 Allison Engineering Department Reports No. 2381.
Second Quarterly Technical Progress Report
Continuation of 3-KW Stirling Cycle Solar Power System Program
(1 August through 31 October 1961)
Dated 14 November 1961.
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Project Engineer: D. S. Monson, Phone CHapel 4-1511, Extension 4866

APPROVED: T. F. Nagey
Director of Research

25

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TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIST OF TABLES

ALLISON

INTRODUCTION

This technical report describes the second quarter activity (1 August through 31 October 1961) on Phase II of the three-phase 3-kw Stirling Cycle Solar Power Systems Program. Phase I, which was conducted from September 1959 through December 1960 under Contract No. AF33(616)-6771, consisted of the design and experimental evaluation of a research model of the advanced Stirling engine, applied research of solar concentrators, heat receivers, thermal energy storage devices, radiators, orientation devices, system controls, seals, and studies preparatory to the design of a deployable concentrator orbital experiment.

Phase II of the program was initiated 1 May 1961 and is to cover a period of 10 months. This phase of the program is directed toward improving the performance of the research advanced Stirling engine to ensure attainment of a net output of 3 kw of electrical power at a minimum brake thermal efficiency of 30 percent in the flight-type engine design. Phase II consists of three basic tasks: (1) engine component investigations, (2) Model PD-46 research engine performance improvement, and (3) flight-type engine design and partial fabrication. The first quarter activity (1 May through 31 July 1961) was reported in Allison Engineering Department Report No. 2236, dated 4 August 1961.

I. OBJECTIVES FOR THE REPORTING PERIOD

The work to be accomplished during the second quarter of the contract period (1 August through 31 October 1961) included: (1) continuation of PD-46 engine problem elimination and performance tests, (2) continuation of power piston seal development program, (3) evaluation of heat exchanger circuit pressure losses under pulsating flow conditions, (4) shakedown ground tests and flight tests of zero "g" lubrication test rig, and (5) completion of the design of the Model PD-67 lightweight engine components. The scope of the work scheduled for the second quarter on these phases of the program was as follows:

PD-46 PROBLEM ELIMINATION TEST PROGRAM

The following tests were planned for the second quarter:

1. Completion of engine control system evaluation
2. More accurate engine heat balance determination, utilizing a water-cooled shroud to evaluate the cylinder heat radiation losses
3. Determination of the thermodynamic phase angle
4. Determination of the working fluid pressure at all critical points in the heat exchanger circuit
5. Evaluation of the effect of reduced transfer piston seal leakage
6. Evaluation of the effect of reduced power piston seal leakage. A run with an O-ring seal, resulting in virtually zero leakage, was planned as a part of this investigation
7. Tests with hydrogen as the working fluid were planned, provided safety studies indicated such a test to be feasible

POWER PISTON SEAL DEVELOPMENT PROGRAM

Three new ring configurations were scheduled for test in the seal test rig in an attempt to overcome difficulties previously encountered. These difficulties consisted of excessive wear, rotation of individual members of the assemblies relative to one another, and extrusion of the rings through the end cuts of the mating rings. The configurations to be tested were as follows:

1. Glass-filled Teflon seal (Koppers K-30) with decreased gap clearance and increased back-up spring loading. This design is similar to the configuration which was quite successful when run in the PD-46 engine. It was anticipated that testing of this assembly would establish better correlation between engine and rig testing.

2. Three-and four-piece ring assemblies of Rulon A material with steel back-up rings with lap end cuts were scheduled for the second test during the quarter, it was thought that no antirotation device would be required with these rings.

3. The third configuration is a three-piece assembly consisting of two sealing rings with interlocking steps to prevent relative rotation. The end cuts are in the thin section of each ring and the unsupported section of the adjacent ring is in the thick section, thereby reducing the tendency to extrude through the end cut gap.

ZERO "G" LUBRICATION TEST

Completion of all ground testing and flight testing in the ASD weightless research aircraft was scheduled for this report period.

THERMODYNAMIC AND FLUID DYNAMIC COMPONENT TESTS

Single Heater Tube Pressure Drop Tests

Evaluation of the effect of smooth approach tube entrances was scheduled for the early portion of the quarter. No additional tests were planned.

Unit Heat Exchanger Assembly Flow Test

Shakedown running of the test rig and completion of the proposed test program was scheduled for the first two weeks of the quarter.

ADVANCED LIGHTWEIGHT ENGINE DESIGN

Completion of the design of the major components of the PD-67 engine was scheduled for the second quarter. Any changes indicated to be required as a result of the component, PD-46, and zero "g" tests were to be incorporated in to this design program.

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II. PROGRAM PROGRESS SUMMARY

The present status of the program is summarized in this section of this report. Detailed discussion of the various phases of the program is presented in later sections.

TECHNICAL STATUS

PD-46 Test Program

The following tests have been conducted during the second quarter of the program:

1. Tests to evalute further the performance of the automatic load control system

2. Tests to obtain more accurate performance data, with particular emphasis on heat balance, temperature ratio, pressure ratio, and gas pressure-volume phase angle

3. Calibration with white metal power piston seal, to evaluate over-all performance, phase angle, and pressure and temperature ratios

4. Recalibration with white metal power piston seal after repair of crack in displacer piston

Seal Development Program

An endurance test of 214 hours' duration was conducted with a glass-filled Teflon (Koppers K-30) three-piece ring assembly. Three other configurations were subjected to leakage and friction evaluation tests.

Zero "G" Lubrication Test

The ground tests in process at the start of the quarter were completed, and all scheduled flight testing was conducted during the quarter. All work on this phase of the program has been completed.

Thermodynamic and Fluid Dynamic Component Tests

Single Heater Tube Pressure Drop Tests

Tests, conducted during the first month of the quarter to evaluate the effect of smooth approach entrance conditions at the heater tube ends completed all work planned for this phase of the program.

Unit Heat Exchanger Assembly Flow Test

All scheduled testing was completed during this quarter including pressure drop measurements under pulsating flow conditions with the regenerator cartridge in three different positions and with and without blended entrances to the heater and cooler tubes.

Advanced Lightweight Engine Design (Model PD-67)

Present status of the major components is as follows:

1. Cylinder Head Assembly. The design layout and dimensional stack drawings are completed. Detail drawings of the components are in process. This design was approved by ASD during the quarter (Letter of 25 October 1961 from Paul E. Wieser, ASD contracting Officer).

2. Drive Mechanism. The design layout was completed during this quarter. The dimensional stack drawing is now in process.

3. Crankcase. Design of a new lightweight crankcase is in process. A design study of the required modifications to the PD-46 crankcase to permit its use with the PD-67 lightweight head assembly and drive mechanism is also in process.

MAJOR ACCOMPLISHMENTS

PD-46 Test Program

Installation of a water jacketed shroud around the engine NaK manifold to measure heat rejection losses, and a more precise calibration of all critical instrumentation has resulted in an increase in calibration data accuracy. Recent data obtained with this refined instrumentation indicate that the performance data presented in the first quarterly report were in error. The heat rejection from the NaK manifold, which is now being measured, differs from the estimated value previously used. In addition, the wattmeter calibration curve which was previously used has been found to be in error.

Analysis of data obtained during the second quarter indicates an electrical power output of 2690 watts with Teflon power piston rings, and 3140 watts with a white metal power piston seal at design conditions. In both cases the engine brake thermal efficiency was approximately 22 percent. At reduced speed (66 percent of design speed) the efficiency was approximately 28 percent for the Teflon ring configuration.

Seal Development Test

Shakedown of the automatic controlling system was completed during the first week of the quarter. This equipment permits continuous operation of the test rig on a 24-hour-per-day basis with only periodic checking by the test personnel.

By careful analysis of the rig power input for various configurations tested during this period, it has been possible to develop a procedure for separating pumping power and friction power requirements. This will be of value in the

future in evaluating new seal materials and configurations.

Zero "G" Lubrication Test

The results of the zero "g" flight test program indicate that some positive means of circulating the lubricant within the crankcase should be provided. This concept is being incorporated in the lightweight crankcase design.

Thermodynamic and Fluid Dynamic Component Tests

Single Heater Tube Pressure Drop Test

Results of these tests indicated a reduction in heater tube pressure loss when the tube ends were blended to provide a smooth approach entrance condition. The heater and cooler tubes of the PD-46 engine have been modified accordingly.

Unit Heat Exchanger Test

Pressure drop measurements under pulsating flow conditions indicate that the regenerator cartridge may be moved closer to the heater or cooler with negligible effect on pressure drop. This leads to the conclusion that no excessive flow restriction exists in the transition spaces between the regenerator and the heater or cooler with the present configuration.

Advanced Lightweight Engine Design

Design studies regarding the extent of required modifications to the present PD-46 crankcase to permit its use with the new PD-67 cylinder head, drive mechanism, and lubrication concept indicate that such a program is quite feasible. This approach would permit separate evaluation of the major PD-67 components in a shorter time than would be required if a new lightweight crankcase were fabricated.

PROBLEMS ENCOUNTERED

PD-46 Test Program

After the first calibration with the white metal power piston seal, a crack was detected in the dome of the displacer piston which necessitated repair of the displacer piston and a rerun of the previous calibration.

Difficulty has been encountered in the attempt to measure the thermodynamic phase angle accurately (phase angle between maximum pressure and minimum volume). Several modifications of the instrumentation have been made, but satisfactory results have not yet been obtained.

Seal Development Program

During the first test of this quarter several delays were encountered due to malfunctioning of the automatic controlling system. Corrective action was taken, and the equipment is now working satisfactorily.

Zero "G" Lubrication Test

Analysis of the movie films obtained during the one "g" ground tests and the zero "g" flight tests shows that little correlation exists in the lubricant splash pattern when tested under these different conditions. It may be concluded from these films that ground tests are not adequate to determine the amount of lubricant required or the splash pattern to be expected under zero "g" conditions.

Thermodynamic and Fluid Dynamic Component Tests

Difficulties were encountered in attempting to obtain accurate pressure measurements under pulsating flow conditions in the single heater tube pressure drop and unit heat exchanger assembly flow tests. Under conditions of pulsating pressure, any volume in the pressure transducer or connecting lines will distort the normal pressure reading. This volume was minimized

by using connecting lines as short as possible. Because of this difficulty it is felt that the results should be considered only as qualitative indications.

FUTURE PLANS

PD-46 Test Program

Tests which will be conducted during the next quarter include the following:

1. Power piston seal leakage evaluation. A test will be run with an O-ring seal in the white metal seal piston which was used during the second quarter. This configuration, which will result in virtually zero leakage, will aid in evaluating the effect of power piston leakage on engine performance.

2. Gas density tests. A white metal seal power piston will be reinstalled and tests will be conducted using helium, nitrogen, and a helium argon mixture as working fluids. These tests will aid in the evaluation of flow losses and the effect of specific heat ratio on engine performance.

The program to develop a satisfactory technique for the measurement of pressure phase angle will be continued during the next quarter.

Seal Development Program

The configuration, which is now undergoing evaluation tests to obtain the relative magnitude of friction and leakage, consists of a piston with three 0.180-in. grooves with each groove containing two outer step-cut Rulon A rings and two back-up springs.

Three additional tests of this nature are planned. They will include tests of Rulon R-5 material with improved end cuts and investigations of lighter ring loading.

At the completion of the evaluation tests, the best configuration will be placed on endurance testing.

Zero "G" Lubrication Test

No future work is planned under the present contract. It is recommended, however, that any future flight lubrication systems be subjected to zero "g" test flights before being incorporated in an orbital or space vehicle.

Thermodynamic and Fluid Dynamic Component Tests

No future testing is scheduled at the present time.

Advanced Lightweight Engine Design

It is currently planned to release the Model PD-67 cylinder head assembly design for fabrication. The future design plans of the other major engine components are currently being re-evaluated commensurate with over-all program objectives subsequently to be worked out with the ASD Laboratory.

III. PD-46 PROBLEM ELIMINATION TEST PROGRAM

Development testing has continued in an effort to achieve the performance goals of the PD-46 Stirling engine. It is desirable to achieve these goals so that necessary modifications can be incorporated in the flight-type PD-67 Stirling engine.

The history of all testing conducted under this contract is summarized in Table I. The following tests were conducted during the second quarter:

Run No. 4.	This run was initiated during the first quarter. However, the automatic control test was conducted during the first week of the second quarter and is described in this report.
Run No. 5.	This test was conducted with refined instrumentation to obtain more accurate performance data.
Run No. 6.	This test was conducted to evaluate performance with the white metal power piston seal (previous build-ups incorporated a power piston with three Koppers K-30 Teflon ring assemblies).
Run No. 7.	This was a rerun of Run No. 6 after repair of a crack in the displacer piston dome.

AUTOMATIC LOAD CONTROL SYSTEM TEST (RUN NO. 4)

The first test made during this report period was a continuation of the control tests which were initiated during the first quarter. During this testing the engine was subjected to rapid changes in load, working fluid pressure, and engine speed, during which its mechanical performance was completely satisfactory at all times.

TABLE I

Test History — Current Contract

Run	Date	Time	Objective	Results
1	6/13/61	0	Base line calibration	NaK leak at bellows while heating loop. Cylinder head damaged beyond repair
2	6/23/61	53 min	Base line calibration	Questionable data due to oil contamination - defective shaft seal
3	7/12/61 through 7/17/61	9 hr 25 min	Base line calibration Heat transfer Windage test Flow loss	Useful data obtained — Test terminated during flow loss test due to heavily scored displacer piston
4	7/25/61 through 8/3/61	10 hr 2 min	Recalibration Flow loss Automatic control	Useful data obtained. Control functioned properly. Questionable heat input data
5	8/24/61 through 8/29/61	10 hr 45 min	Obtain refined heat balance Determine temp ratio, pressure ratio, and phase angle	Accurate heat balance obtained. N_t 28% at 60% speed, N_t 22.5% at 100% speed. Instrumentation inadequate for temperature and phase angle test
6	10/13/61 through 10/18/61	11 hr 56 min	Evaluation of clearance (white metal) seal Determine phase angle, temperature & pressure ratio	3% increase in output. Disassembly revealed excessive leakage due to cracked displacer piston. Pressure and temperature ratio near design. Phase angle instrumentation inadequate
7	10/26/61	2 hr 3 min	Repeat Run 6 with repaired displacer piston and improved phase angle instrumentation. Check effect of reduced dead volume	12% increase in power output. Phase angle instrumentation inadequate. Removed to install zero leakage power piston
8	Anticipated 11/6/61		Repeat Run 7 with O ring piston seal Improved phase angle instrumentation Further reduced dead volume	

Total Running Time -	Current Contract	45 hr	4 min
" " "	Original Contract	76 hr	17 min
	Total	121 hr	21 min

These control tests were conducted after completion of the control panel repairs and stand wiring modification as described in the first quarterly report. This control panel maintains constant engine speed by varying the power input to a parasite load resistor. The original design concept provided a parasite resistor of 500 watts' capacity, assuming that the engine was to be operated at or near full power at all times. However, the "breadboard" control components are capable of diverting up to 13 amp at 170 volts (2200 watts) to the parasite resistor. The parasite load resistor was therefore selected at 13 ohms with 13 amperes capacity to provide maximum possible control. As the amount of parasite load was controlled by varying the portion of each cycle that the parasite load connected to the alternator, the resistor chosen caused distortion of the voltage at the control panel. At relatively large parasite loads, this distortion affected both the voltage and frequency sensing circuits and, as a result, the alternator voltage and engine speed. At relatively low parasite loads, voltage and speed control were reasonably satisfactory except for superimposed oscillations. Simultaneous recordings of alternator voltage, alternator frequency (engine speed), and parasite load current obtained during the test are shown in Figure 1.

The initial useful load was 1125 watts, which was increased to 1425 watts in one step. As the useful load increase (300 watts) was more than the initial parasite load, the parasite load was completely removed, as indicated by zero current to the parasite load in Figure 1. Because the power output of the engine was less than the 1425 watts required by the applied useful load, the engine speed was reduced, resulting in a decrease in voltage; and a new condition of equilibrium was established.

The "breadboard" control is being modified to include damping in both frequency and voltage control. Also, the parasite load resistor has been changed to 27 ohms at 6.5 amperes capacity to reduce the voltage distortion. Additional testing may be conducted during the next quarter.

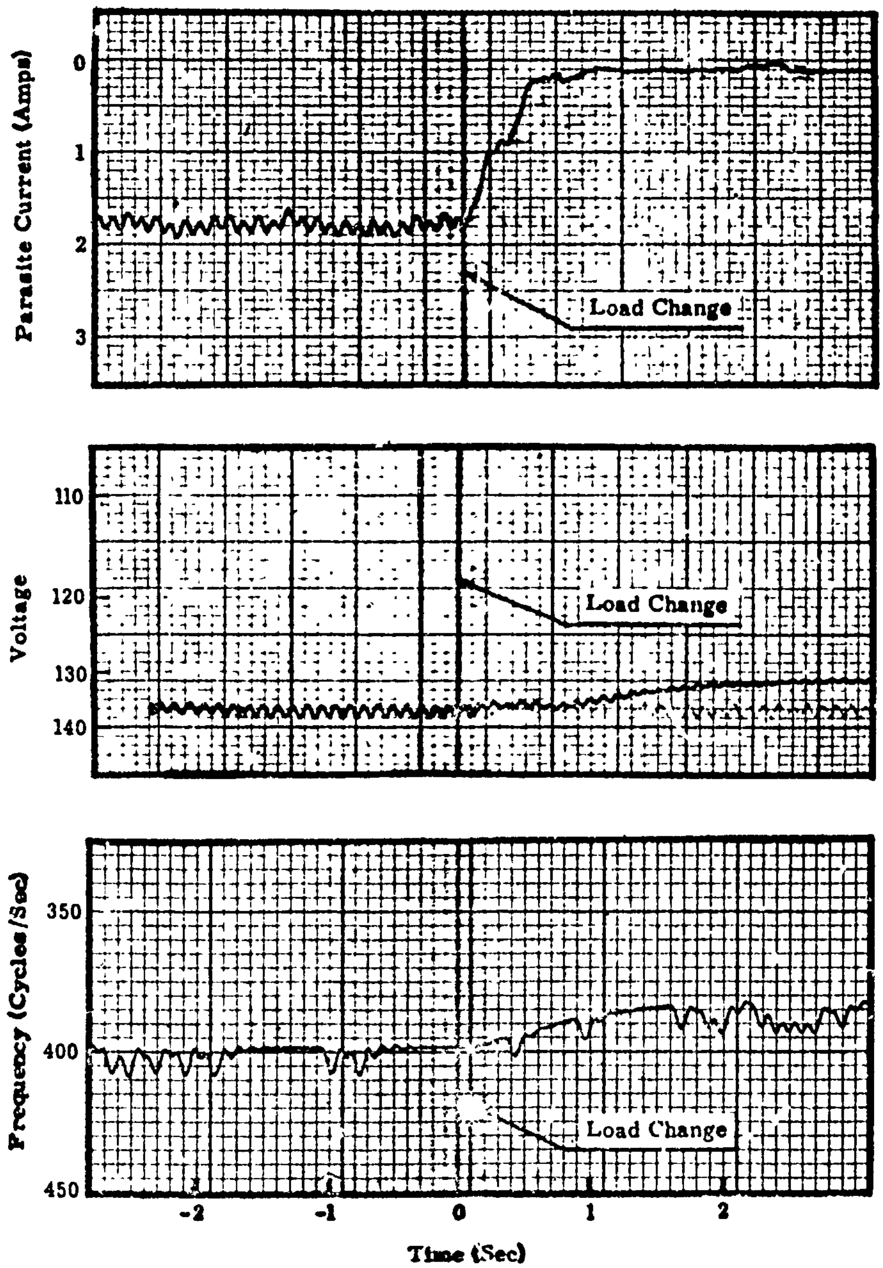

Figure 1. Automatic Load Control System Response

PERFORMANCE CALIBRATION WITH REFINED INSTRUMENTATION (RUN NO. 5)

At the completion of the control system test, it was necessary to modify the engine to measure the pressure phase angle, expansion zone gas temperature, and obtain a more accurate heat balance.

Pressure Phase Angle

To measure the pressure phase angle within the engine, a pressure transducer was installed in the compression space cavity and a power piston location indicator was placed on one generator rotor. See Figures 2 and 3.

Expansion Zone Gas Temperature

It was desired to measure the expansion zone gas temperature to establish whether or not the heater section was performing satisfactorily. A 1/8-in. tube was installed through the NaK cavity into the expansion space so that a thermocouple could be inserted. Because of the small clearance between the displacer piston and the dome of the expansion space, it was necessary to locate the tip of the thermocouple just inside the dome wall.

Heat Balance Improvement

During previous tests, considerable difficulty was encountered in obtaining a complete heat balance since the radiation and convection heat losses from the engine could not be accurately determined. It was deemed necessary to evaluate these losses to allow a more specific analysis of data and isolate loss areas more completely. Several steps were taken to measure all heat additions, removals, and losses.

A complete and accurate calibration was made of all the instrumentation including pressure gauges, thermocouples, voltmeters, ammeters, and wattmeters, the tachometer, and flowmeters. Complete calibration curves were compiled over the range encountered during engine testing. Mixer

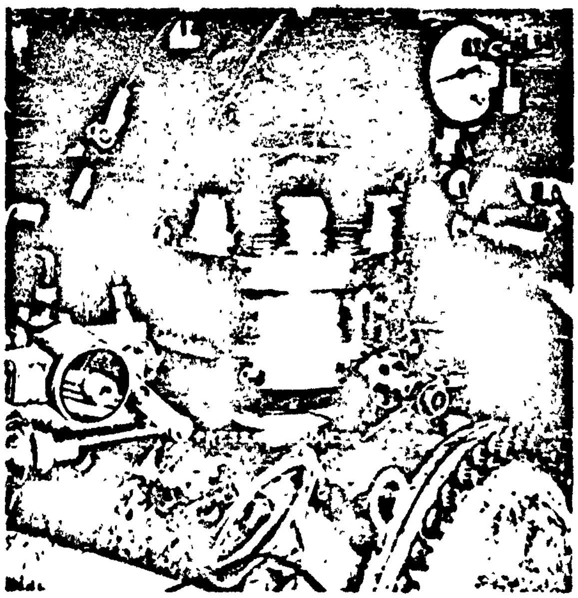

Figure 2. Locat[illegible] Pressure Transducer

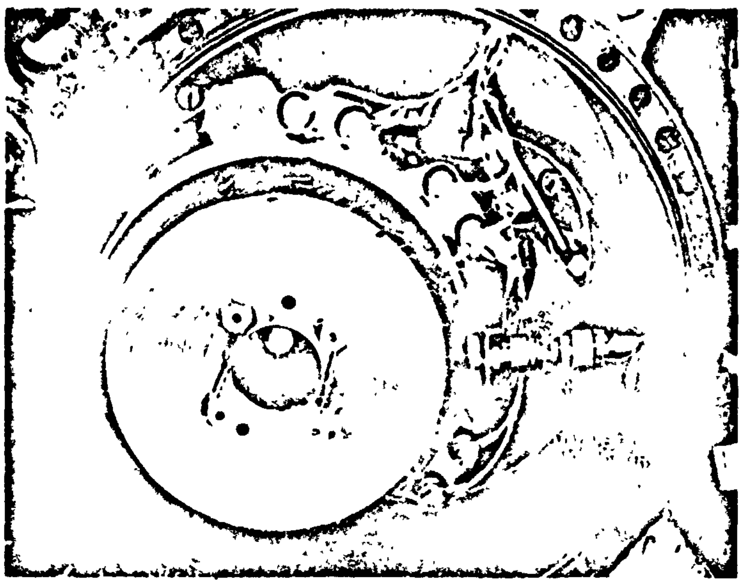

Figure 3. Magnetic Indicator [illegible] Power Piston Location

baffle plates were installed in the NaK inlet and outlet lines to the engine to ensure proper mixing of the NaK flow before it passes the thermocouples.

The entire engine was thoroughly insulated to reduce convective heat losses from the crankcase. A water calorimeter was installed around the head section to facilitate measurement of the radiant heat loss from the NaK cavity.

As a result of these modifications, heat balances were obtained in which the unaccounted for wattage was less than 10. This agreement is well within the limitations of the measuring devices.

WHITE METAL PISTON TEST (RUNS NO. 6 AND 7)

The previous test (Run No. 5) included three Koppers K-30 glass-filled Teflon ring assemblies in the power piston. It was considered necessary to evaluate the white metal piston in the engine further, since this is the configuration most likely to be used in the flight-type PD-67 engine. A complete engine teardown was conducted to ensure that all components were in good condition.

Engine Teardown Inspection

This engine inspection followed 30 hours and 12 minutes of engine running, during which the mechanical operation of the engine was satisfactory. At the time of shutdown no specific mechanical trouble was apparent. Inspection revealed several minor areas which require further investigation and testing. With the exception of the following variations, the engine was in completely satisfactory condition.

Piston Shaft Seals

Slight oil leakage was found to have occurred through both the displacer piston shaft seal and the power piston shaft seal. The leakage through the power piston shaft seal amounted to approximately 20 cc of oil, while that through the displacer shaft seal was an indeterminate amount that had adhered to the piston and gas passages. This was partially caused by the re-use of

the damaged displacer shaft seal from the previous build-up because of the unavailability of a new seal.

Slight scoring of the shafts in the area of the guide bushings was also noted. Indications are that these shafts will require a higher surface hardness for extended operation. All future shafts will be hardened by carburizing the seal and bushing area.

Regenerators

The regenerators were removed from the cylinder head for complete inspection. Each cartridge weight was compared with that before testing, and the results were found to be inconclusive in that most cartridges showed weight gains of approximately 0.05 gram while two indicated a weight loss of 0.06 gram each. Since these two weights were obtained on different scales (one at Harrison Radiator Division and the other at the Allison Division) it was not possible to recheck the initial weight.

Visual inspection revealed a deposit on the screen mesh on the cold side of each regenerator (see Figure 4). This deposit is believed to be a combination of oil and wear residue from the Teflon K-30 piston rings. A flow test at teardown indicated that the pressure drop through the side arm circuit increased approximately 7 percent when compared with the check made at Build-up 7. This deposit was removed by ultrasonic cleaning, followed by conventional cleaning processes. Damage to the screen was observed on the hot side of the regenerator where heater tubes coincided with regenerator end plate holes (see Figure 5). This is felt to have been caused by bombardment of small foreign particles in the woking fluid against the screen which [illegible] been weakened by the 1100°F temperature. This problem indicates the need for extreme cleanliness during assembly.

Generators

Slight generator rub occurred, but it was not severe enough to cause trouble. The high spots on the stator have been stoned down to prevent further rubbing.

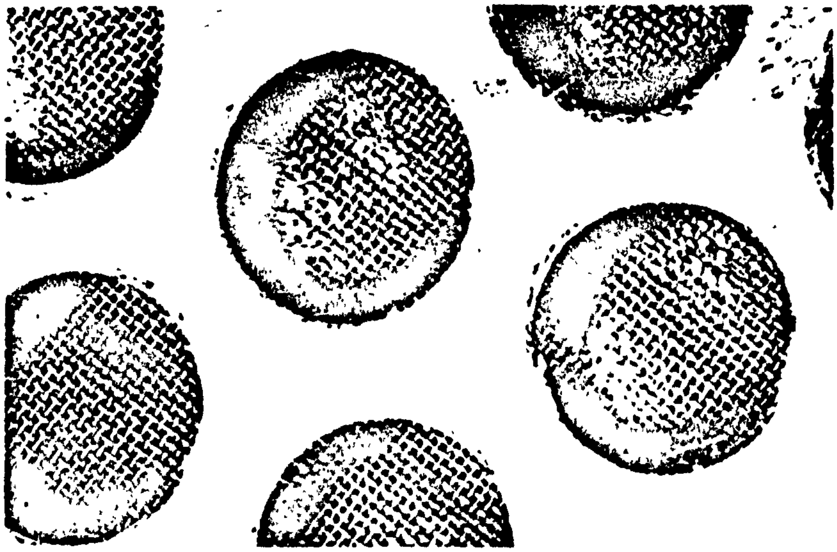

Figure 4. Deposits on Screen Mesh on Cold Side of Regenerator

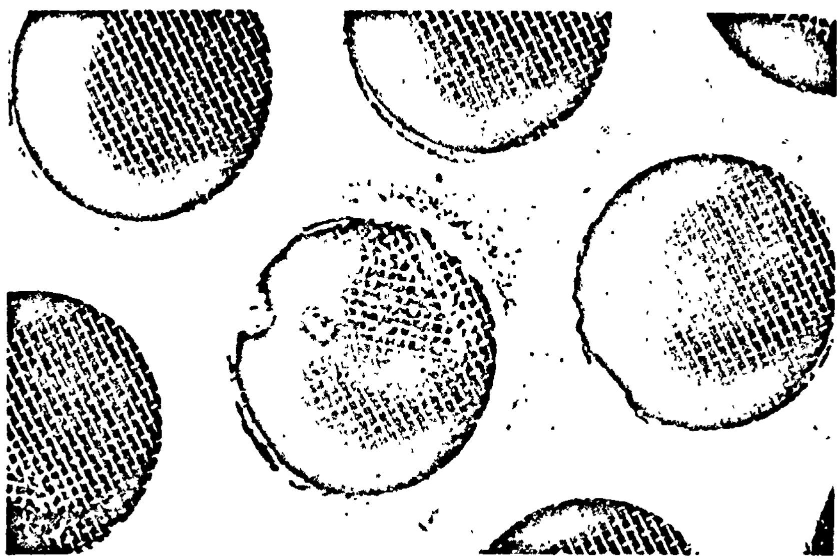

Figure 5. Screen Mesh Damage on Hot Side of Regenerator

Two pieces of tape, used to hold wires during fabrication, had been loosened by oil (see Figure 6). Delco Remy Division has assured Allison that it was an oversight that the tape had not been removed previously and that, so long as the wires were rigid, the tape is not required.

Engine Build-Up

Three white metal pistons were fabricated and sized for use during the latest build-up in the event one should be damaged, worn, or modified in any way that would prohibit its re-use. One piston was prepared by spraying lead-tin alloy on as suggested by N. V. Philips. MoS_2 powder was sprayed onto the piston concurrently, so that it was dispersed throughout the plating. This piston was installed at the build-up and performed satisfactorily.

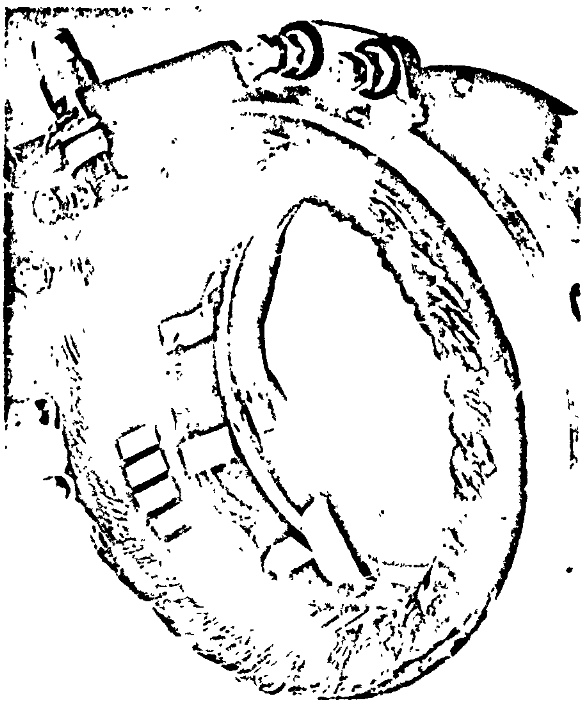

Figure 6. Tape Straps Loosened by Oil

Two other pistons were prepared by an Allison Division electroplating process with a different concentration of MoS_2 placed in the plating solution for each piston. One plated piston has a MoS_2 concentration similar to that of the Dutch sprayed piston, while the second piston has a very low concentration.

The white metal seal on the displacer piston was also replated to reduce its clearance so that blow-by would be reduced. Because the expansion zone gas temperature measurements obtained on the last test (Run No. 5) proved to be unsatisfactory because the thermocouple location resulted in readings that were not the true gas temperature, but more nearly the temperature of the cylinder dome wall. The displacer piston was modified for the next test by placing a recess well in the dome to allow the extension of the thermocouple 1/4 inch into the gas stream.

The entrances to all the gas passages in the three heat exchangers were rounded to reduce flow losses. Previous tests had indicated that flow losses may be a source of loss within the engine.

Test Run No. 6

After 11 hours and 56 minutes of running the pistons were removed and inspected. Although the performance improvement that had been expected was not realized, the white metal seal piston appears to have operated satisfactorily. The two pistons, after test, are shown in Figure 7. Dimensional checks revealed that the power piston diametral clearance at the end of this run was only 0.0006 inch. This clearance is greater while running, since the high pressure within the cylinder causes the cylinder wall to move away from the piston.

The displacer piston white metal seal clearance had increased from 0.0000/0.0002 inch at build-up to 0.0019/0.0033 inch at teardown and was slightly elliptical, with the maximum diameter being parallel to the crankshafts.

It was found that the weld around the thermocouple well in the displacer piston had cracked because of grinding off too much of the weld. This resulted in

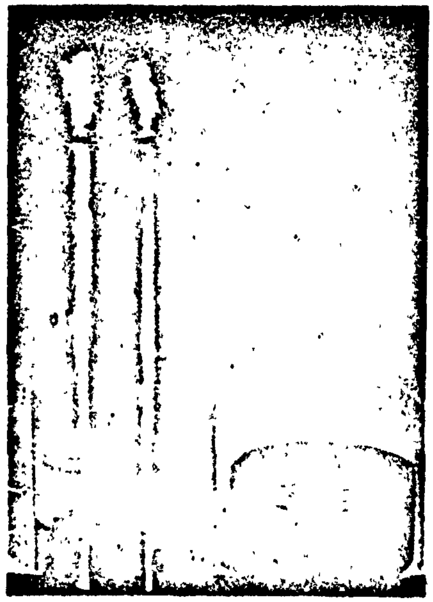

Figure 7. The Two Pistons After 11 Hours and 56 Minutes of Testing (Run No. 6)

gas leakage into and out of the piston while running. A new weld bead was placed over the old weld to prevent further leakage. A 0.016 diameter wire was placed in the 0.021 diameter vent tube in the displacer piston to further reduce the flow into and out of the piston.

The displacer piston was then regrooved to provide a better seal. No rework was done on the power piston. The engine was then reassembled for Run No. 7.

Test Run No. 7

After two hours and three minutes of running, the pistons were again removed and examined. No additional wear was experienced on either piston. The two pistons after this run are shown in Figure 8. Both white metal seal bands were in excellent condition.

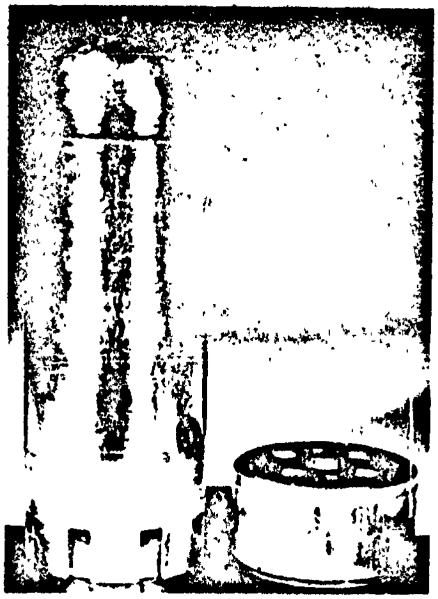

Figure 8. The Two Pistons After Run No. 7

The temperature ratio measurements indicated that design values of gas temperature are being achieved in both the heater and cooler section.

Pressure phase angle measurements still have not been acceptable. The location of both the transducer and the piston location indicator appear to be satisfactory, but the signals require additional amplification to produce sufficient signal strength to provide an accurate record.

Although the design objectives have not been obtained, the engine appears to be operating extremely well from a mechanical standpoint.

PD-46 PERFORMANCE ANALYSIS

Several theories have been explored for improving the performance. These theories are basically concerned with improving the heat transfer coefficients in the heater and cooler. Although the initial measurement of gas temperature in the expansion space resulted in thermocouple readings of 1150°F at the design point, an unfavorable condition existed in the location of the thermocouple which may have affected the accuracy of the reading. The thermocouple was installed flush with the cylinder wall instead of protruding into the hot space. This permitted the thermocouple to be heated by wall conduction and radiation without much chance of the gas removing this heat effectively. Steps were later taken to correct this condition. The fact that the heat input was considerably below the theoretical value and the heat rejection was slightly above its theoretical value led to the suspicion that the heater temperature was low. Figure 9 shows that decreasing heater temperature has exactly this characteristic—that is, the heat input decreases rapidly with decreasing temperature and heat rejection increases slowly. With the placing of the actual values on this curve, it can be seen that a heater temperature of approximately 900°F would give the measured performance.

A low heater gas temperature could be caused by one or more of the following:

1. Low heat transfer coefficient on the NaK side

2. High thermal resistance in the tube wall or any coating in conjunction with it

3. Low heat transfer coefficient on the gas side

The NaK side heat transfer has been tested by varying the NaK flow rate and observing the change in an over-all heat transfer coefficient. This test indicated that the heat transfer was not limited by the NaK side. Several uncertainties in the evaluation of this test do not eliminate consideration of this possibility, and the design of PD-67 will incorporate features to improve the flow condition of the NaK over the tubes and higher NaK velocities will be used.

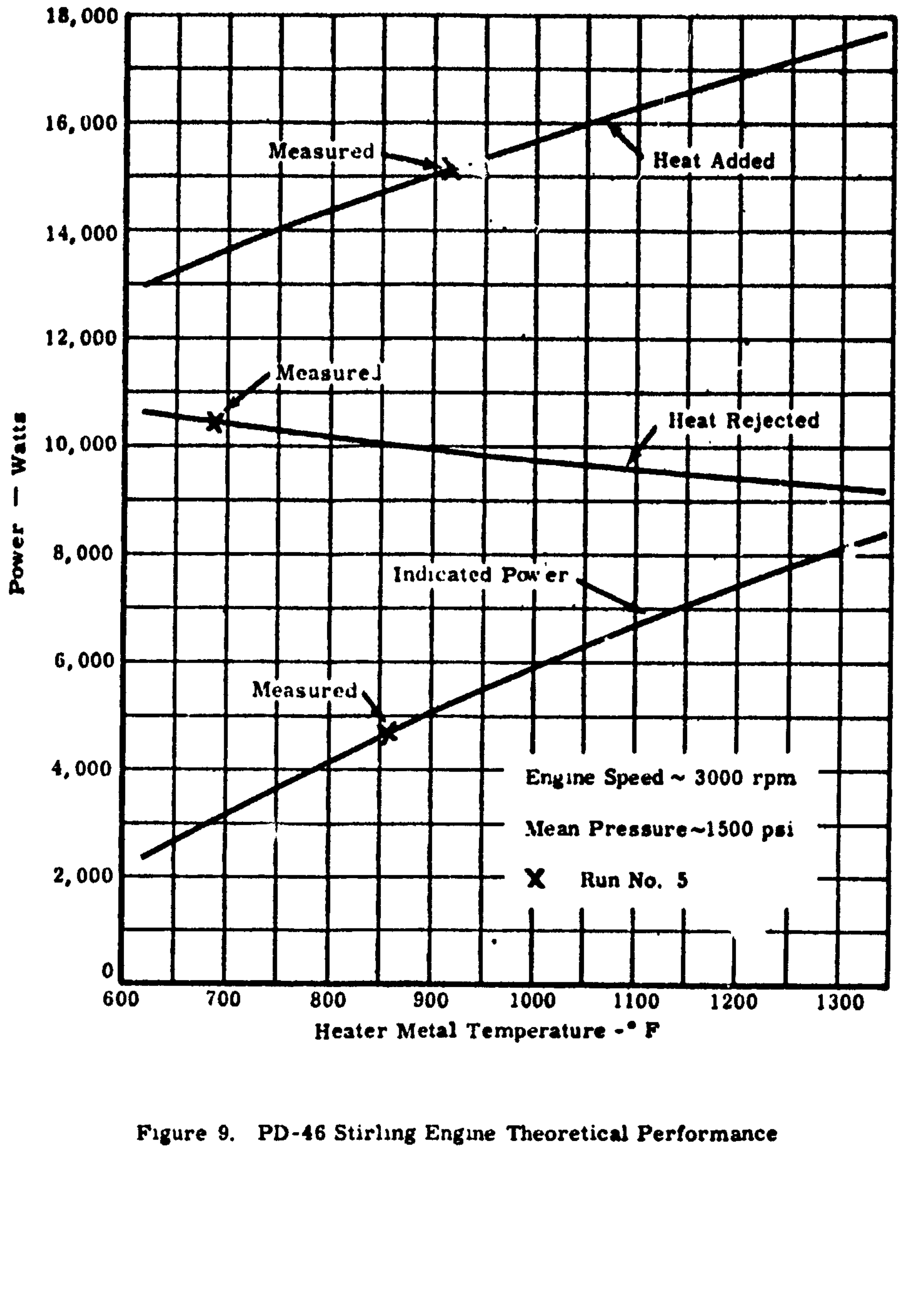

Figure 9. PD-46 Stirling Engine Theoretical Performance

The temperature drop through the heater tube wall has been calculated to be 17°F. A layer of low thermal conductivity material either on the inside or outside of the tubes could increase this temperature drop. Although this possibility is considered to be slight, it should be kept in mind.

The data that were used to estimate the gas side heat transfer coefficient were taken from Reference 1 for the laminar flow regime and Reference 2 for turbulent flow.* Since transition normally occurs between Reynolds numbers of 2000 and 10,000, the data in the transition region were obtained by fairing the data from both sources, as shown in Figure 10. The mean Reynolds number and tube l/d ratio for PD-46 is indicated on the curve.

As is apparent from Figure 10, the mean Reynolds number is in the transition region where the heat transfer coefficient is quite uncertain. In fact, if the most pessimistic value corresponding to the laminar flow is used, this could easily explain the discrepancy between theoretical and actual performance. If one considers the disturbed flow condition at the tube entrances, it would hardly seem likely that laminar flow could exist at Reynolds numbers of 8000. On the other hand, it must be remembered that a high frequency flow oscillation, with complete stoppage and reversal of flow taking place, could conceivably alter this situation.

Many investigators (such as those in Reference 3) have studied pulsating flow, but not with stoppage and reversal. The authors of Reference 3 have been consulted as to their views on this subject and have agreed that the possibility does exist that transition could be delayed in such a flow situation.

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*1. Norris and Streid, Transactions ASME, August 1940, p. 525.

2. McAdams, Heat Transmission, 3rd edition, p. 226.

3. Siegel and Perlmutter, Heat Transfer for Pulsating Laminar Duct Flow, ASME 61 - SA - 28.

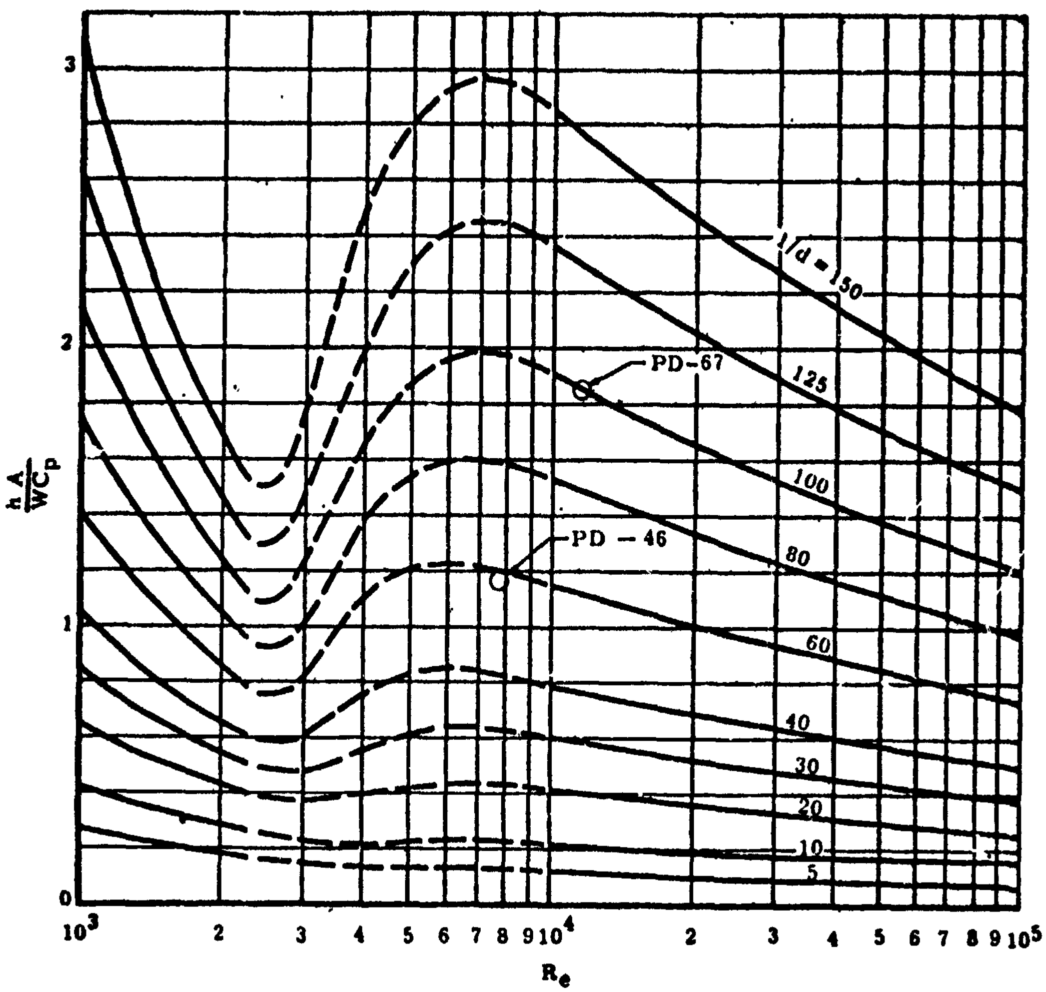


Figure 10. Heat Transfer Coefficient vs Reynolds Number

During past tests using nitrogen as a working gas, the mean Reynolds number was 60,000, which is definitely turbulent. Although the flow losses were much higher with nitrogen, when allowance is made for this, it is found that the actual performance is closer to theoretical than with helium. This can be seen in Figure 11, where the power difference is plotted against engine speed. It is felt that improved performance could have been caused by the Reynolds number being high enough to ensure turbulent flow. The remainder of the loss with nitrogen remains to be explained.

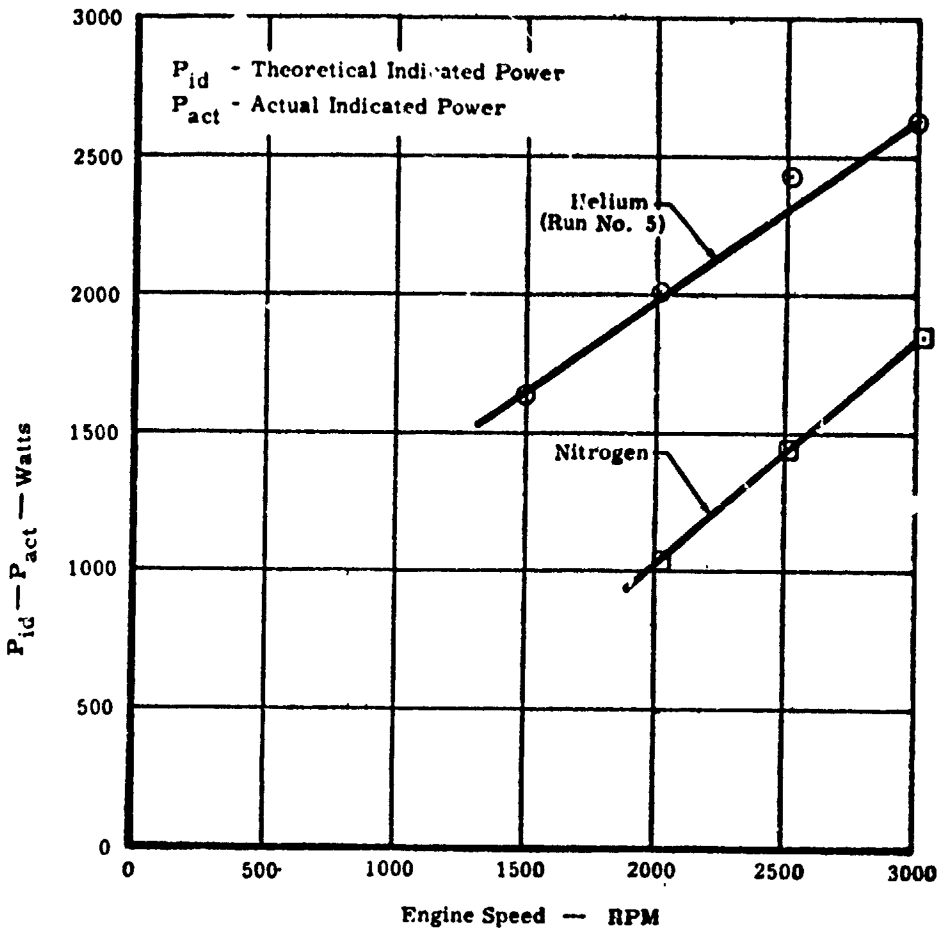


Figure 11. PD-46 Performance with Helium and Nitrogen Working Fluids

If delayed transition is causing difficulty, it is relatively simple to increase the Reynolds number in future designs. For the PD-67, a mean Reynolds number of 12,000 and a tube l/d of 100 has been tentatively selected to improve the gas side heat transfer coefficient.

It should also be noted that a high gas temperature in the engine cooler could produce the same characteristics as a low heater gas temperature. This possibility is considered slight, however, since the measured cooler gas temperature is fairly close to the theoretical value.

Test Results

Three engine performance test runs have been completed during this quarter. In the first test (Run No. 5) the power piston was fitted with three Teflon rings as seals. In the second test (Run No. 6) the white metal power piston was used. For the third test (Run No. 7) the same piston used in Run No. 6 was utilized and the dead volume of the engine was considerably reduced.

Performance data from Run No. 5 is presented in Figures 12 and 13. The mechanical losses shown in these curves were those obtained by measurement of the heat rejected from the oil cooler and subtracting known values for the heat losses from the generators. This heat, therefore, includes bearing and shaft seal friction and windage from generators and linkage. The friction heat from the power pistón seal was probably partially measured in the crankcase and the rest in the head water cooler, as the cylinder wall in this area ran hotter than both the crankcase and the head water cooler. By adding the mechanical losses to the brake engine output, values of "indicated" output (or cycle output) were obtained. The indicated output was also obtained by subtracting the cycle heat rejected from the heat added. This number is not considered to be as accurate as the values obtained by the engine output method, but surprisingly good agreement was obtained in the two numbers when the temperatures in the engine were allowed to stabilize for a long time. At the design point, for instance, there was only 10 watts difference in the two values. A breakdown of the heat balance is shown in Figure 14. Another feature that was added for these tests was a calorimeter surrounding the head

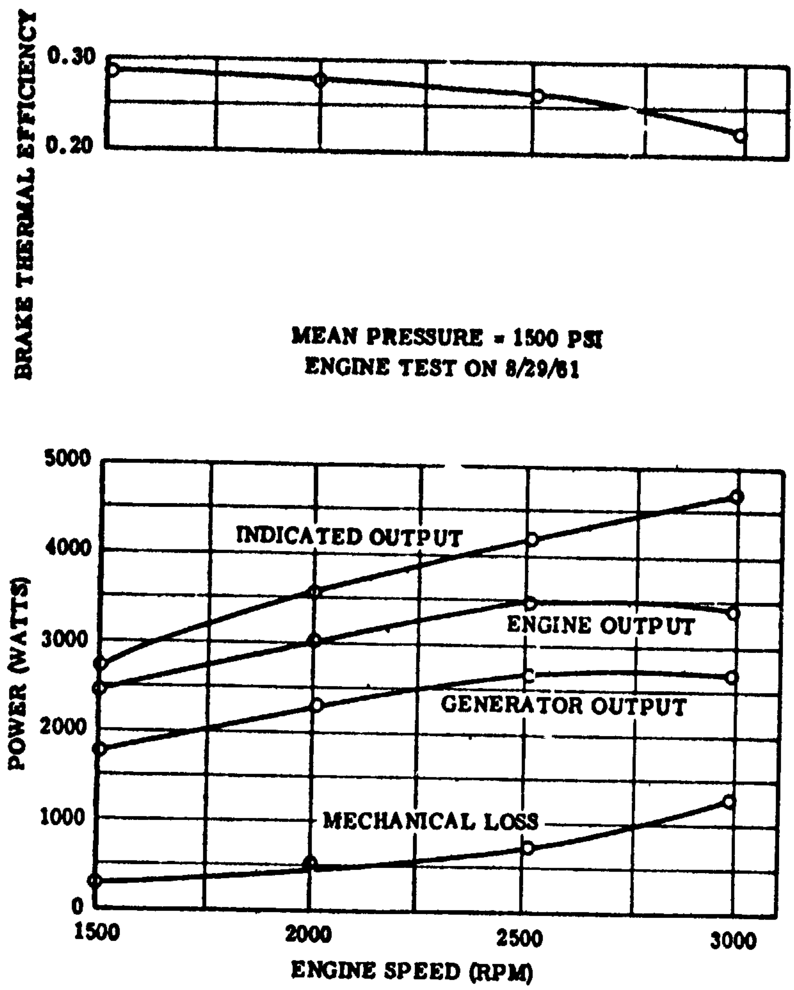


Figure 12. Stirling Engine Test Results (Run No. 5) — Engine Speed vs Power and Brake Thermal Efficiency

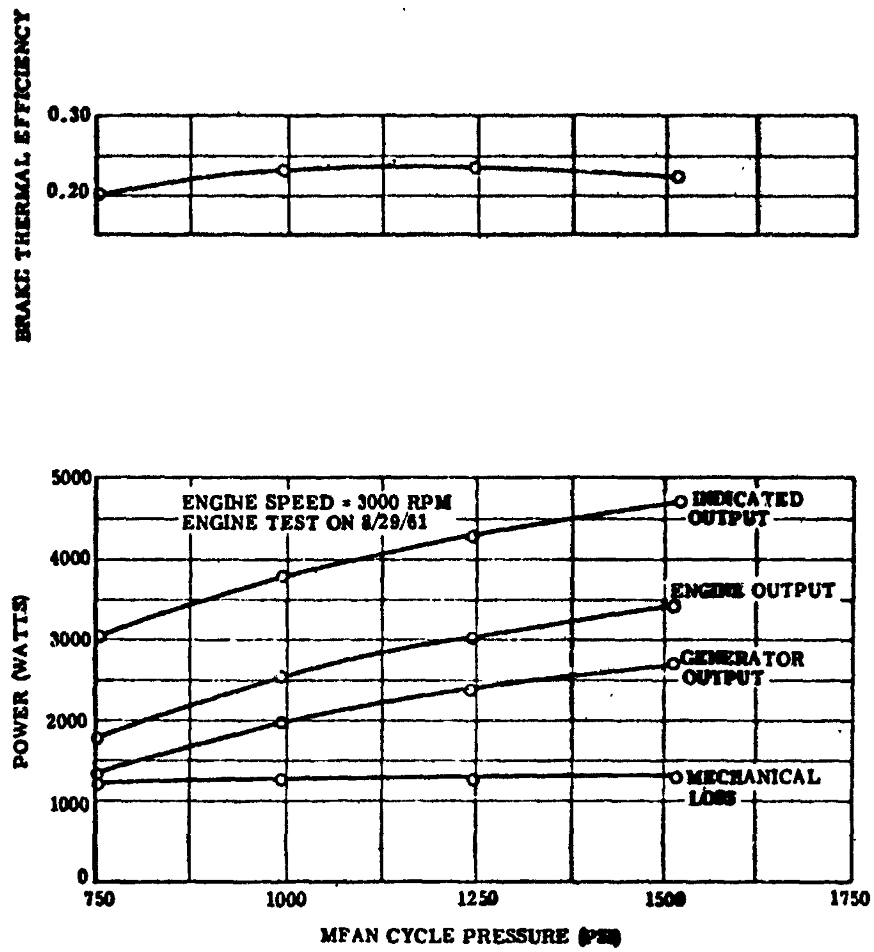


Figure 13. Stirling Engine Test Results (Run No. 5) — Mean Cycle Pressure vs Power and Brake Thermal Efficiency

Engine Speed = 3000
Mean Cycle Pressure = 1500 psi

All Values In Watts

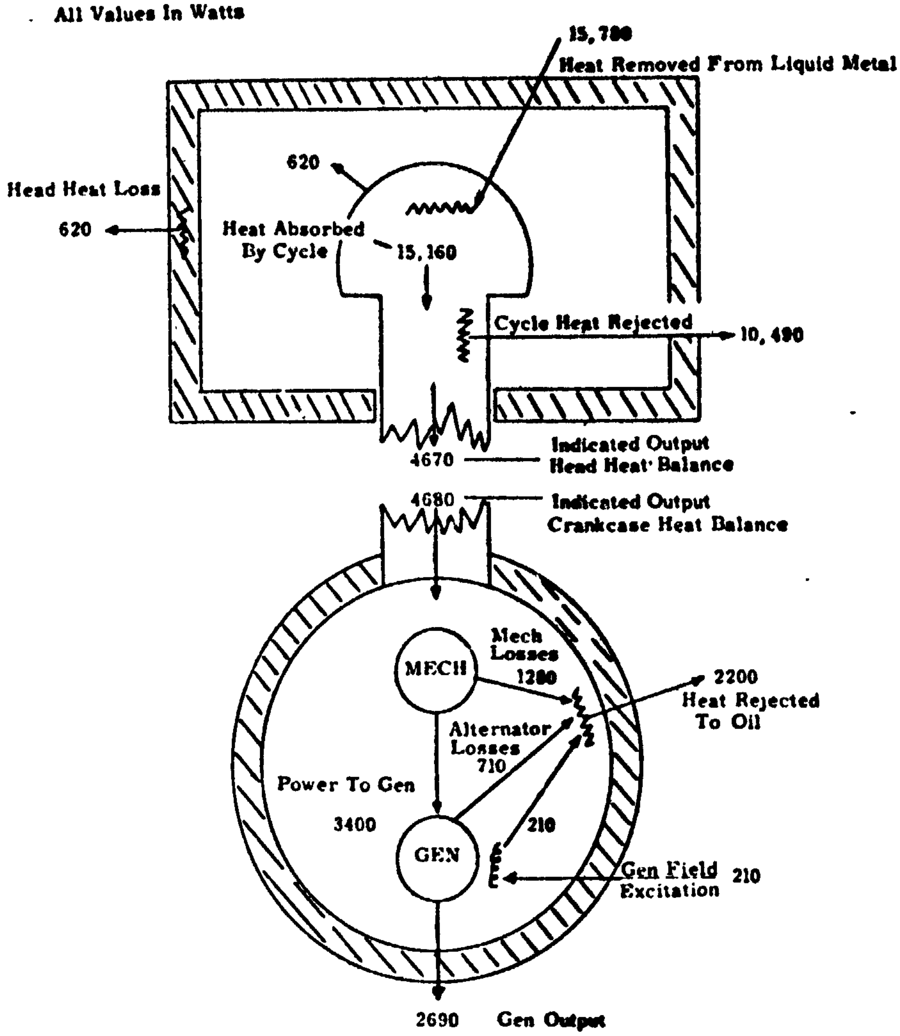


Figure 14. Stirling Engine Heat Balance Diagram

to measure the head heat loss. Water circulating through the calorimeter at near room temperature picks up the heat lost from the head and facilitates its measurement.

This test has shown that the previous estimate of mechanical losses was low. This was caused by the inability to measure the heat loss from the crankcase to the room. When the crankcase was insulated was insulated, this heat loss was eliminated. When the new values were used the indicated power was higher and did not exhibit the characteristic of flattening out at high speeds. In fact, when the difference between theoretical and actual indicated power was calculated, it was found that it varied linearly with speed and pressure, as shown in Figures 15 and 16. The old values are included for comparison.

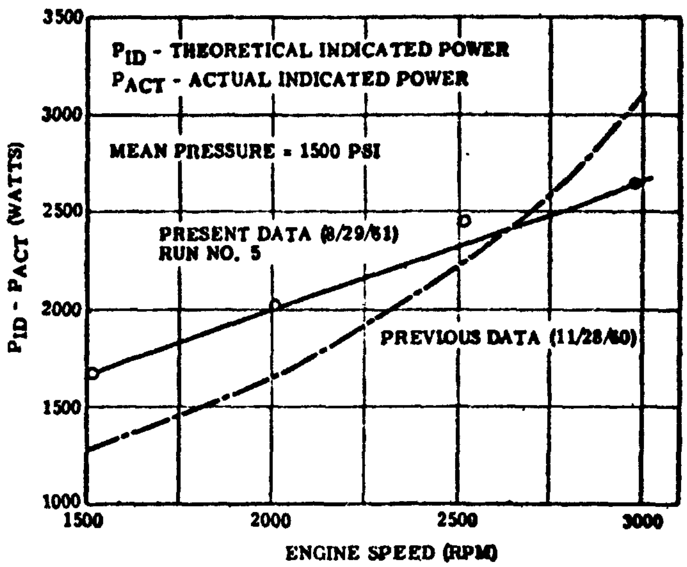


Figure 15. Stirling Engine Test Results

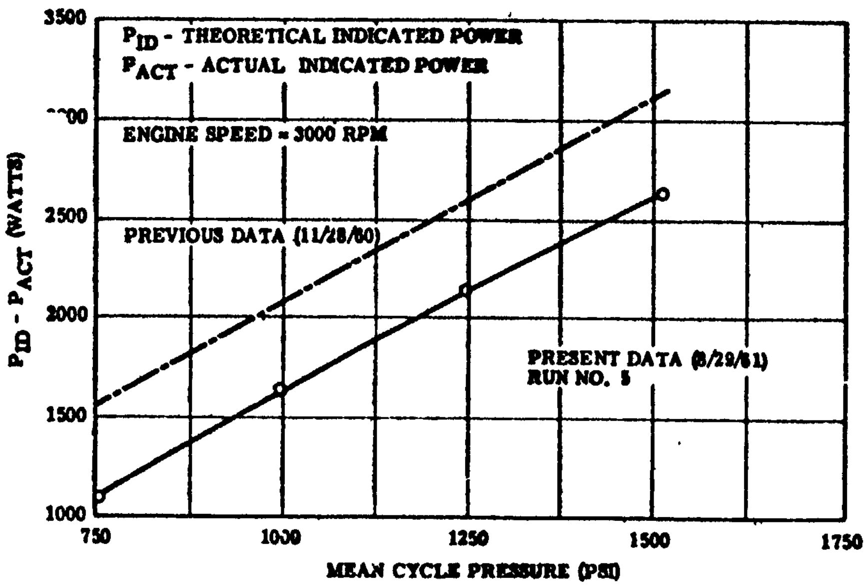


Figure 16. Stirling Engine Test Results

After the testing was concluded, a recheck of the wattmeter indicated no readable shift in the calibration of the instrument.

A continuous monitoring check was made on the NaK ΔT instrumentation by using two laboratory L & N precision potentiometers manned by trained instrumentation personnel. Extremely close correlation was obtained between the stand instrumentation and the precision lab equipment at all steady state points during the entire run.

In Run No. 6 the white metal power piston was installed in the engine. Also, a depression or well was put in the displacer piston in order that the thermocouple could extend farther into the gas stream for more accurate measurement of the expansion space temperature. The performance data from this test are shown in Figures 17 and 18. As can be seen from these plots, the output is slightly higher than that obtained with the ring seal at 3000 rpm and lower at lower speeds. This appears to be a result of lower

seal friction, which helps at high speed, at the expense of increased leakage which has a greater effect at low speed. A post-test inspection of the piston seal leakage indicated, however, that the leakage was less than was experienced previously with the ring seals. Further inspection revealed that a crack had developed in the weld around the depression in the displacer piston used for the thermocouple which explains the indication of higher leakage shown by the test data.

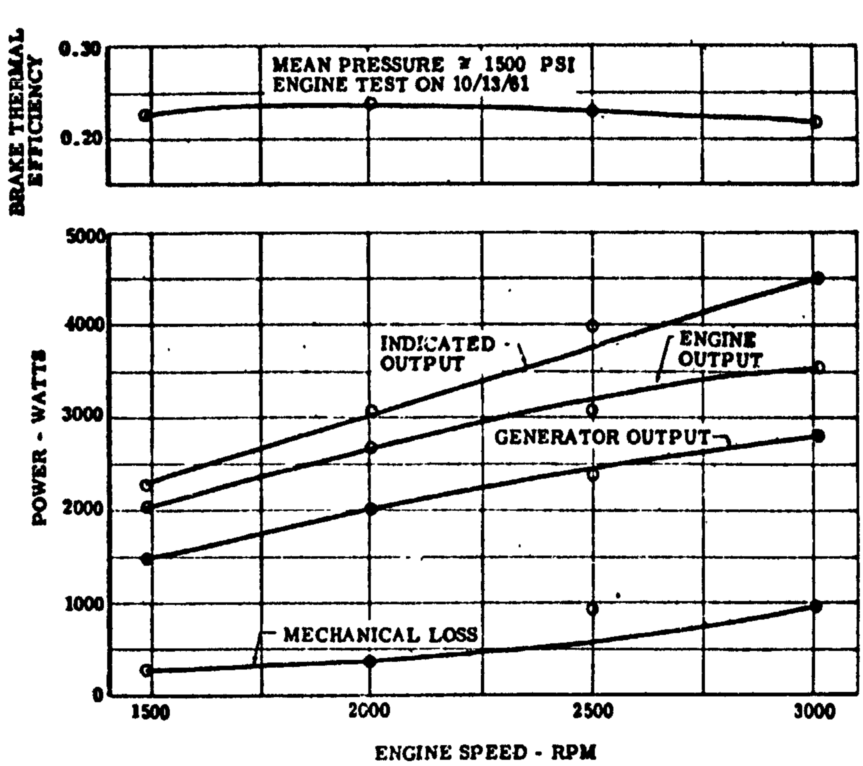


Figure 17. Stirling Engine Test Results (Run No. 6) — Engine Speed vs Power and Brake Thermal Efficiency

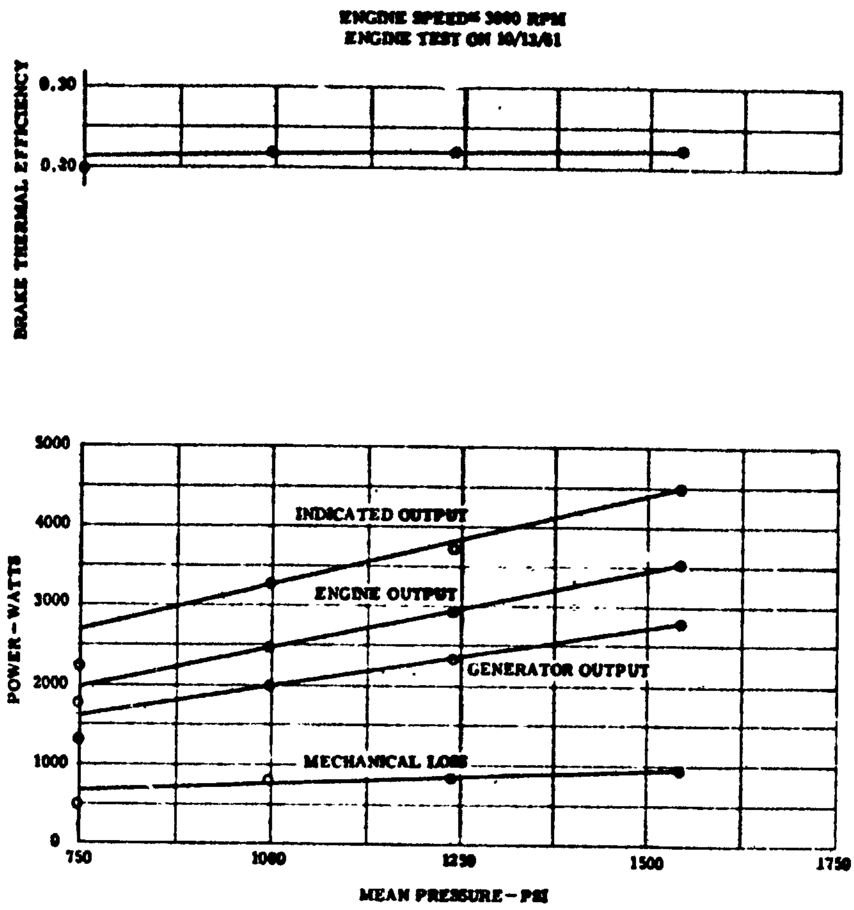


Figure 18. Stirling Engine Test Results (Run No. 6) — Mean Cycle Pressure vs Power and Brake Thermal Efficiency

The next step was to reweld the displacer piston to eliminate the leakage in this area. It was also deemed desirable to eliminate or minimize all other possibilities of gas leakage in and out of dead spaces. To accomplish this, the cross-sectional area of the displacer vent tube was reduced by inserting a wire, and in addition the maximum and minimum pressure check valves were removed. The valve used to vent the head to the buffer space was filled with epoxy and then drilled out to minimize the volume, and the "snubber" used to dampen the pressure oscillations to the mean pressure gauge was improved by using a long capillary tube rather than the porous plug. The results of this test (Run No. 7), shown in Figure 19, indicated that considerable improvement was made over Run No. 6 at all speeds, and that the performance was about the same as that of Run No. 5 at low speeds. The improvement made at high speed is encouraging. Since the low-speed performance has not improved over Run No. 5, it is felt that additional improvement can be made by further reduction of leakage. To this end, therefore, a future test is planned to reduce leakage to a minimum with a rubber O-ring seal. This test is solely intended to evaluate the ultimate in seal leakage, and lubrication will be provided at build-up in an amount considered sufficient to permit a short duration test.

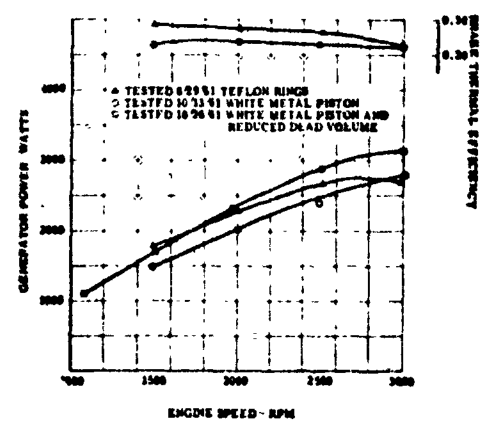


Figure 19. Test Results — Run No. 7

Also included in the last tests was an improved thermocouple position in the expansion space. The theoretical and actual mean temperature in both the expansion and compression spaces is shown in Table II. The good agreement in these numbers indicates that, contrary to the theory previously mentioned, the heater and cooler heat transfer coefficients are quite close to the predicted values.

TABLE II
Temperature Comparison

	Temperature - °F	
	Theoretical	Actual
Expansion Space	1057	1075
Compression Space	208	232

Several attempts were made to measure the phase difference between pressure and volume. This phase angle is very improtant as it is a measure of the actual cycle output. Assume a harmonic variation of volume

$$V = V_o \ (1 - \cos\alpha)$$

where V_o is the displacement and α is crank angle. Assume, also, a harmonic variation of pressure,

$$P = P_{mean} \left[1 + \delta\cos\ (\alpha - \phi)\right]$$

where

$$\delta = \frac{P_{max} - P_{mean}}{P_{mean}}$$

and θ is the phase difference between peak pressure and minimum volume (top dead center of piston). The work per cycle then becomes

$$W_k = \oint pdV = \oint P_{mean} \left[1 + \delta \cos (\alpha - \theta)\right] \frac{V_0}{2} \sin \alpha \, d\alpha$$

$$= \frac{\pi}{2} \delta P_{mean} \; V_0 \sin \theta$$

It can be seen from this equation that if the maximum and mean pressure is measured, all that remains to be measured is the phase angle, θ, in order to find the cycle output.

The first attempt to measure the phase angle was with the use of a probe between a transducer and the working space. Excessively large phase angles were measured in this manner and the reason was at first believed to be a phase shift in the probe. The engine cooler housing was modified to accept the transducer directly below the cooler tubes. Large phase angles were again obtained. The electrical equipment was then closely inspected and the galvonometer used in the recording oscillograph was found to be resonant at 100 cps and to give an appreciable phase shift at 50 cps.

Another attempt was made to eliminate the electrical phase shift by using a phase meter and an oscilloscope. The phase meter was found to be unsatisfactory for the type of signals generated by the transducer and the magnetic pickup used to identify top dead center of the piston. The oscilloscope, although fairly satisfactory, suffered somewhat in poor readability and reproducibility. For the next tests a different oscilloscope will be tried which, it is hoped, will improve this situation. Additional attempts are also being made to calibrate the electrical phase shift in the recording oscillograph.

It is felt that the information obtained from the phase angle measurements will be useful in evaluating configuration changes and to observe the variation of phase angle with engine speed. This information should give insight into the degree of gas leakage, possible heat transfer difficulties, and the possibility of acoustic resonance effects.

IV. ENGINE COMPONENT INVESTIGATION

SEAL DEVELOPMENT PROGRAM

During the second quarter, one seal assembly, a three-piece glass-filled Teflon ring seal (Koppers K-30) completed a 214-hour endurance test. Three seal configurations—two five-groove pistons with two-piece ring seals made of Rulon A and a three-piece ring seal made of Koppers K-30—completed configuration evaluation tests. A fourth seal configuration, a four-piece ring seal made of Rulon A, is now being tested. This testing is discussed in the following paragraphs.

Three-Piece Ring Seal Assembly

This seal consisted of a steel piston with three ring grooves. Incorporated into each groove was a three-piece ring assembly. Each ring assembly consisted of two outer square cut rings made of Koppers K-30 material (glass-filled Teflon) and one steel back-up spring. The end cuts in the outer rings were indexed 180° from each other and the cut in the back-up ring was indexed 90° from the outer ring cuts. During operation no direct leakage paths existed through the ring, and the seal depended primarily upon the seal clearance in the groove and the integrity of the sliding seal on the cylinder wall.

In view of results from previous testing, these rings were fitted with an average room temperature clearance of 0.002 inch. Also, a spring producing 32.8 psi loading was used. A groove clearance of 0.002 inch was selected to provide sufficient clearance for preventing the rings from locking in the grooves and still provide a good seal at operating temperature. The 32.8-psi loading spring was used for two reasons: (1) to provide similarity between this test and the PD-46 running, and (2) to provide sufficient loading to prevent ring floating.

This seal was run in a Type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55, had a surface roughness of five micro-inch RMS, and was round within 0.0002 inch and straight within 0.0002 inch for the used length. The entire testing of this seal, a total of 213.75 hours, was accomplished at a speed of 3000 rpm and with a head temperature above 200°F. Other test parameters are shown in Figure 20.

The average performance of the rings with regard to wear is indicated in Figure 20; the detail wear rates are indicated in Table III.

TABLE III
Wear Rate - mg/hr

Ring Position	Test Period—Hours 0-90	0-150	0-214
Top	2.68	5.00	4.79
Center	2.42	2.09	2.24
Bottom	2.06	5.93	4.37

The general tendency of the top and bottom rings to have the largest wear rate was the same as noted in earlier testing. The tendency for increasing rate of wear with time is opposite from earlier testing. These rings produced a wear rate two to three times larger than earlier testing of Rulon A material after the same period of time. This is the same difference as predicted by the material screening program.

The test cylinder displayed an unusually high wear for this test. The bore dimension had increased 0.002 inch on the diameter and heavy scratches were noted. Apparently the glass filler used in the K-30 material has unusually high abrading characteristics.*

*No measurable wear has occurred in PD-46 testing using this seal material. The difference in engine and test rig geometry may be the cause of this discrepancy.

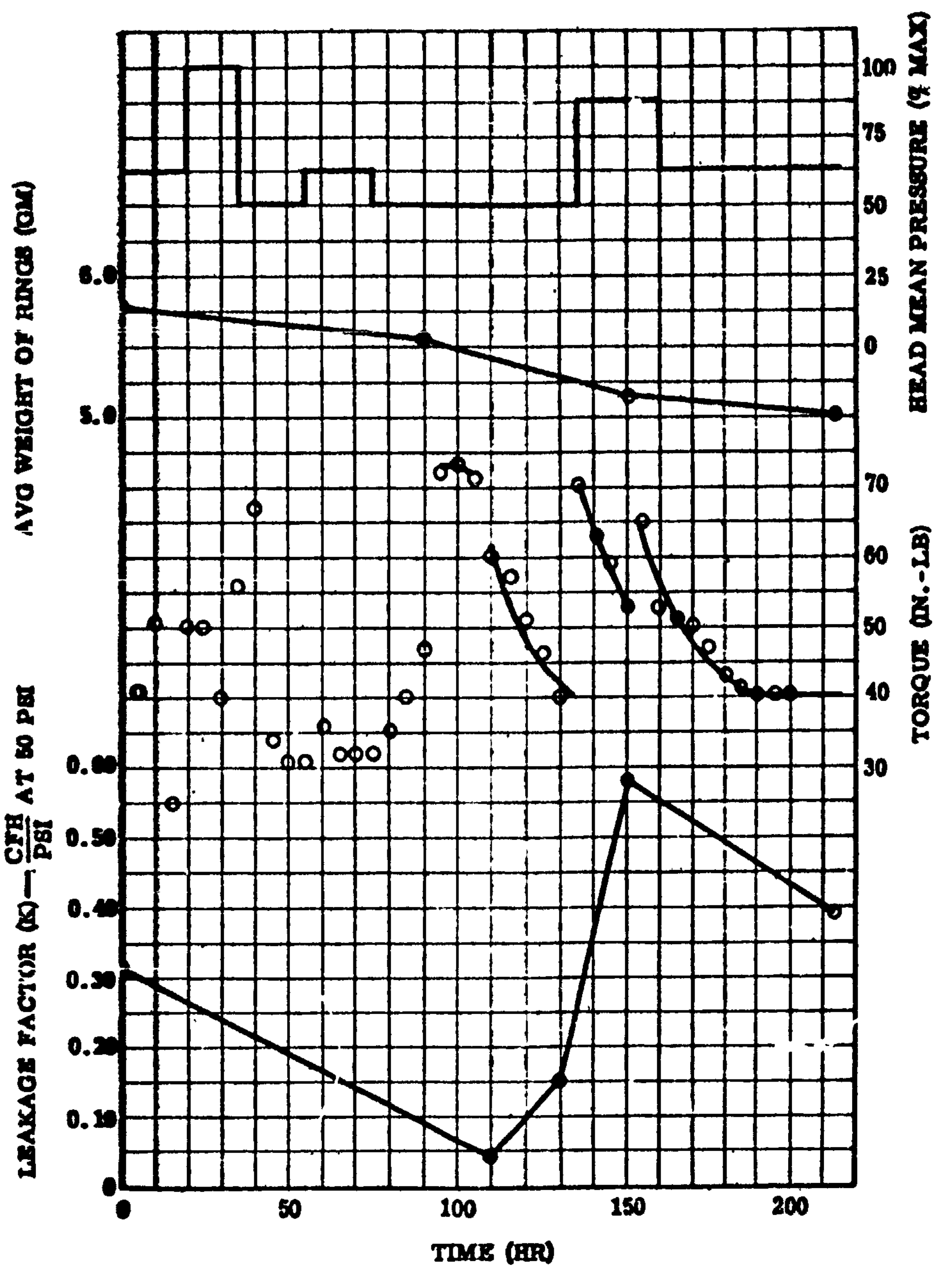


Figure 20. Three-Piece Ring Seal Assembly Testing Results

During the first 90 hours of operation, the test was interrupted on several occasions by malfunctions of the automatic controlling system. Testing after this time was continuous except for teardown for wear measurements. It may be seen from Figure 20 that the torque tended to start very high and decrease with continued running. This characteristic is different from the last test where the torque remained rather constant except for its dependency upon pressure.

This test was discontinued because of excessive leakage, torque, and wear.

At this point it was decided, because of the lack of complete correlation between seal rig testing and PD-46 engine testing, that it would be more advantageous to utilize the rig test to conduct leakage and friction evaluation tests of various seal materials and configurations. The combinations which showed the most promising sealing and friction characteristics would then be subjected to endurance tests at a later date.

Five-Groove Piston with Two-Piece Ring Seal Assemblies

This seal consisted of a steel piston with five 0.090-inch wide ring grooves. Incorporated into each groove was a two-piece ring assembly. Each ring assembly consisted of one outer square cut ring made of Rulon A material and one back-up spring made of precision L-605 material. The end cut of the outer ring was indexed 180° from the back-up ring cut.

For these tests, the rings had an average room temperature groove side clearance of 0.0005 inch and the back-up springs produced 16 psi loading. These two parameters were selected to produce a minimum of leakage and torque, respectively.

This seal was run in a Type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55, had a surface roughness of six micro-inch RMS, and was round within 0.0001 inch and straight within 0.0002 inch.

The cylinder bore diameter was oversize by 0.005 inch and the piston diameter was undersize by 0.010 inch. This test was of relatively short duration and, for this reason, no attempt was made to determine wear rate.

These tests were conducted by installing a single-ring assembly into the piston and increasing mean head pressure while operating at constant rpm. At the completion of this test, a static leak check was taken. This procedure was repeated for two, three, four, five, and zero number of rings. The results from these tests are shown in Figure 21.

As can be seen in Figure 21, the static leakage was excessive at all times.* The leakage rate generally decreased with the number of rings with the exception of five rings for low pressure differentials. As pointed out earlier, the cylinder was oversize by 0.005 inch. This produced an increase in end gap of 0.015 inch. The oversize cylinder, in combination with a 0.010-inch undersize piston, produced a radial clearance which was 0.015 inch larger than necessary. These two dimensions serve to produce a flow area five times that desired and largely account for the excessive static leakage.

The torque characteristics shown in Figure 21 indicate that this configuration was producing a very good dynamic seal. The torque curve with zero rings shows the effect of extremely high leakage, which is obviously significant. With the addition of a single ring, the input torque is very near its minimum value. This is due to effecting a considerable improvement in leakage without an appreciable increase in friction. The minimum torque condition at operating pressure occurred with two rings; the addition of rings above this

*K max = 0.3

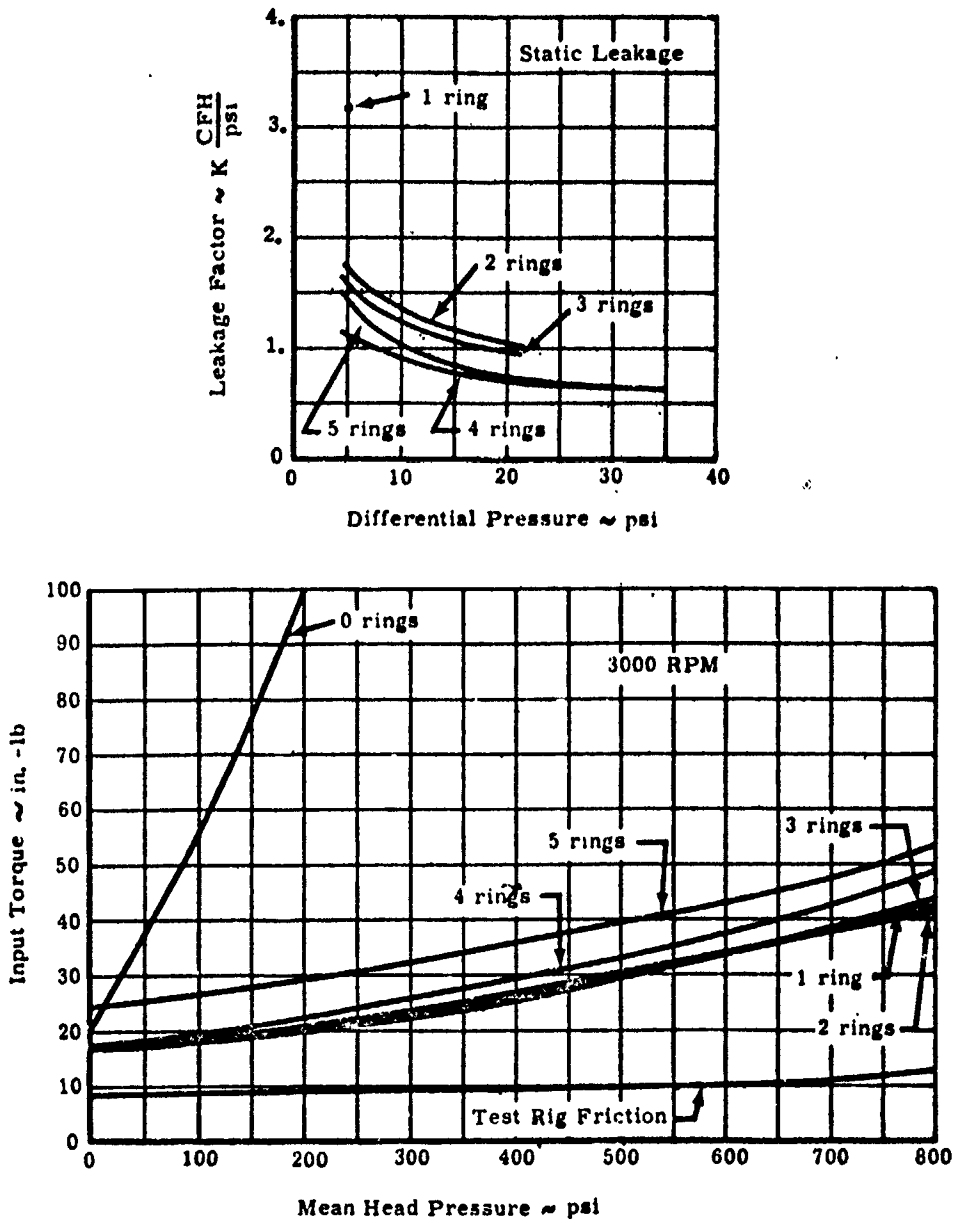


Figure 21. Piston Ring Performance — Rulon "A" Rings with Square End Cuts

point caused an increase in input torque. This is due to the fact that the improvement in sealing is small compared to the increase in friction. From this investigation it would appear that the seal which would produce the largest power output from the engine should contain two or three rings.

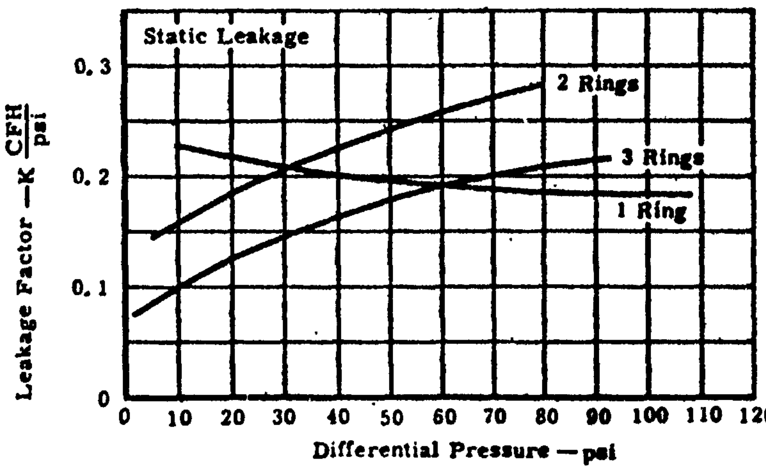
Three-Groove Piston with Two-Piece Ring Seal Assemblies

This seal consisted of a steel piston with three 0.180-inch ring grooves. Incorporated into each groove was a two-piece ring assembly. Each ring assembly consisted of a single outer ring made by Koppers of their K-30 material (glass-filled Teflon) and had a lap cut. Behind the outer ring was a steel back-up spring. The end cuts between the outer and inner rings were indexed 180° from each other. During operation no direct leakage path exists through the ring.

These rings were fitted with an average room temperature clearance of 0.001 inch, and a spring producing 16 psi load was used. These parameters were selected to produce a minimum of leakage and torque, respectively. This ring configuration is identical to that used in previous PD-46 engine testing with the exception that the back-up spring for these tests produced one-half of the loading of the engine configuration.

This seal was run in a Type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55, had a surface roughness of three micro-inch RMS, was round within 0.0001 inch and straight within 0.0001 inch, and was of the correct nominal diameter. This test was of a relatively short duration and, for this reason, no attempt was made to determine wear rate.

These tests were conducted identically with those for the five-groove piston, and the results are shown in Figure 22.



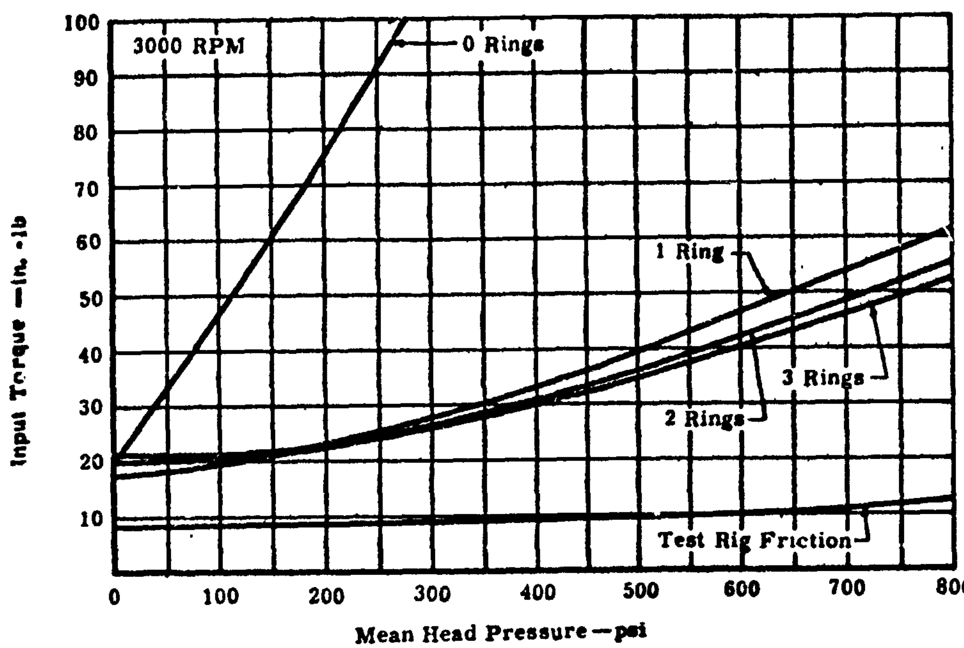


Figure 22. Piston Ring Performance — Koppers K-30 Rings with Lap End Cuts

As can be seen in Figure 22, the static leakage was within the desired maximum value* for all number of rings. For low pressure differentials the leakage rate was as expected—i.e., decreasing leakage with increasing number of rings. Sealing with one ring had a tendency to decrease with increasing pressure and with two or three rings it increased with increasing pressure.

The torque characteristics were similar in nature to the earlier test; however, the torque was higher at high values of working pressure (i.e., 52-62 in.-lb vs 41-53 in.-lb). The fact that this seal produced a higher torque value and that the torque always decreased with increasing number of rings leads one to suspect that this configuration did not produce as fine a dynamic seal as the earlier test configuration although the static leakage was far superior. The effect of smaller piston clearance may be seen for the ringless configurations by comparing Figure 21 with Figure 22.

This seal has a definite tendency for "pumping" at higher pressure; this pumping produced mean pressure differences between the head and buffer zone approaching 200 psi.

<u>Five-Groove Piston with Two-Piece Ring Seal Assemblies</u>

This seal consisted of a steel piston with five 0.040-inch wide ring grooves. Incorporated into each groove was a two-piece ring assembly consisting of one outer step-cut ring made of Rulon A material and one back-up spring. The end cut of the outer ring was indexed 180° from the back-up ring cut.

For these tests, the rings had an average room temperature groove side clearance of 0.0005 inch and a piston cylinder radial clearance of 0.005 inch. The back-up springs produced a ring loading of 16 psi.

*K max = 0.3

This seal was run in a Type 440C stainless steel cylinder which was hardened and tempered to Rockwell C55, had a surface roughness of four micro-inch RMS, and was round and straight within 0.0001 inch.

These tests were conducted by installing a single-ring assembly into the piston and recording torque at constant pressure while varying speed, and at constant speed while varying pressure. At the completion of each test a static leak check was taken. This test procedure was repeated for two, three, four, and five rings. The results of these tests for the same conditions as given for previous tests are shown in Figure 23.

Inasmuch as this seal configuration was run at varying speeds, it became possible to calculate the fraction of ring power requirements which was due to friction and that which was due to leakage. These results are shown in Figure 24. As may be seen from Figure 24, leakage effects are quite significant, amounting to more power than that of friction for one ring. The sum of leakage and friction is, nevertheless, lower for one ring than for the other combinations; that is, friction increases more than leakage decreases for an increasing number of rings.

This ability to separate friction and leakage is a vary strong tool, for now seals may be evaluated as to friction of the material and the sealing integrity of the design.

Material Screening Program

To date, a total of six materials have undergone screening wear testing. These are: Koppers K-30, which has been used for a large part of the PD-46 engine testing; Rulon A, which has been used to a great extent in this program; Rulon B, C, and R-5; and Kel-F.

The results most notable are that Rulon A has 40 percent of the wear rate of K-30, and Rulon R-5 has 20 percent of the wear rate of K-30.

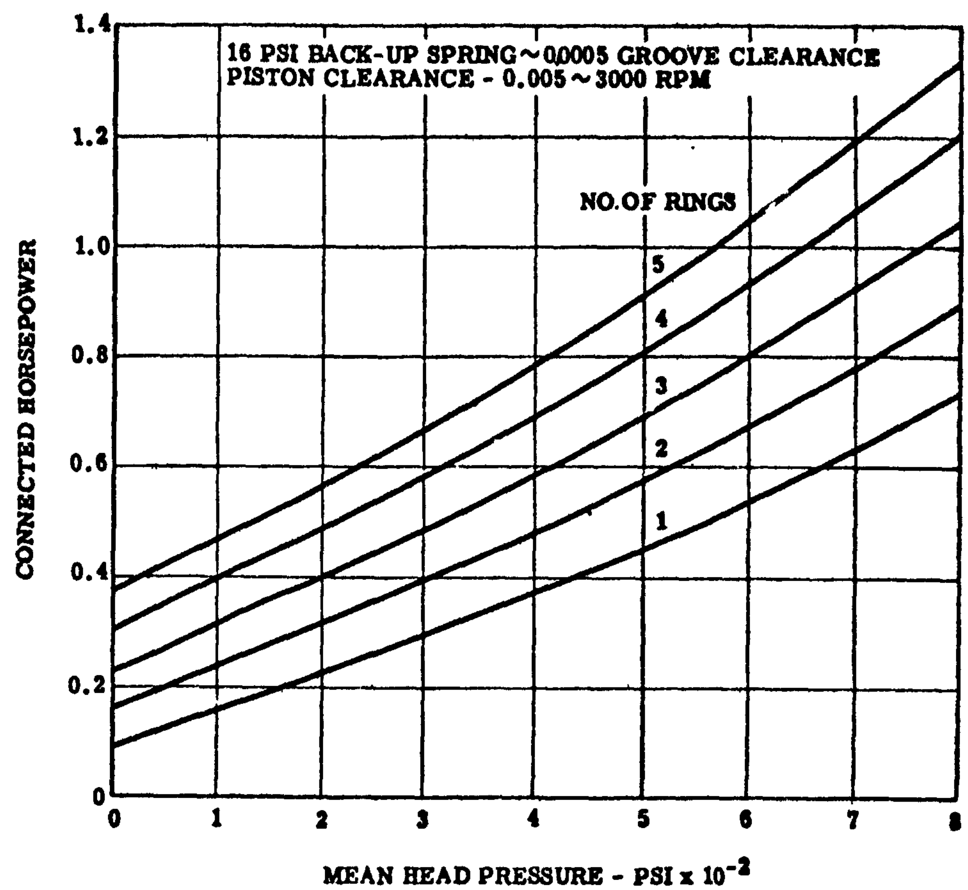


Figure 23. Piston Ring Performance — Rulon "A" Rings with Step Cuts

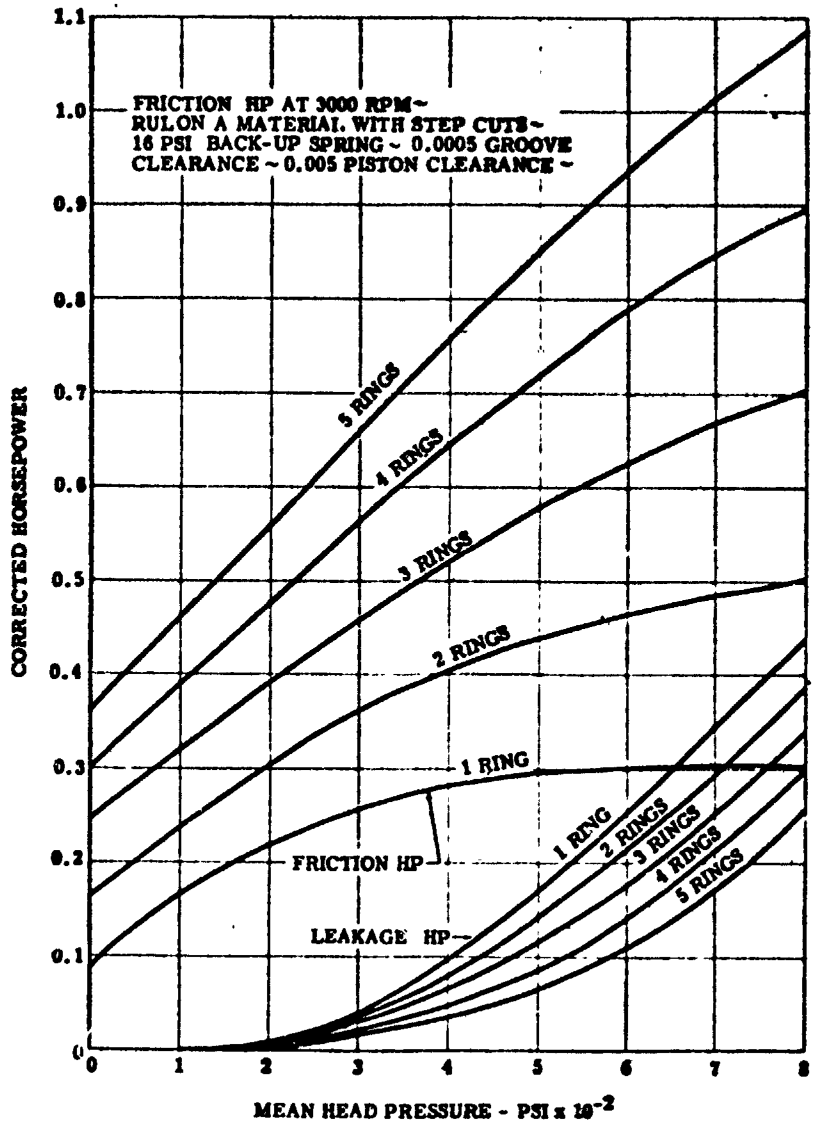


Figure 24. Piston Ring Performance — Power Requirements for Friction and Leakage

ZERO "G" LUBRICATION TEST

Ground testing of the zero "g" test equipment was completed during the first week of the quarter. During this testing, a passage was bored in the plastic baffle to measure the pressure at the oil discharged from the gear teeth. When it was found that no pressure build-up existed in a one "g" environment, it was assumed that the clearance between the gear teeth and baffle was too great to provide a pumping action. Modification of the test rig in an attempt to increase the oil pressure did not seem justified at this time.

The test rig was delivered to Wright-Patterson Air Force Base during the second week of the quarter to await the flight test program in the ASD zero "g" test aircraft. Due to a required overhaul of the KC-135, the tests were included in the C-131 airplane schedule. The Allison test equipment was assembled in one of the WPAFB laboratories and a successful test run was accomplished. The equipment was loaded into the C-131 and checked out with the aircraft power prior to the scheduled take-off. During the first three maneuvers the camera did not fire, apparently due to low voltage. The airplane crew made some adjustments to the airplane power supply and on the fourth maneuver the camera and other equipment operated normally. The flight was aborted at this point because an Air Force passenger observer had become severely ill. Tests were delayed for 12 days due to weather, other flight commitments, and a periodic aircraft inspection.

A photograph taken during this flight (Figure 25) shows Air Force technicians releasing the test capsule as it becomes weightless within the cabin of the USAF C-131 during a zero "g" parabolic trajectory.

On 21 August a successful test flight was made in the C-131, and eight rolls of film were exposed. The film was processed and previewed. Photographically, the film was very good and it revealed quite graphically the zero "g" lubrication patterns within the crankcase. The trajectory flown by the C-131 airplane is shown in Figure 26 and a summation of the film data is presented in Table IV.

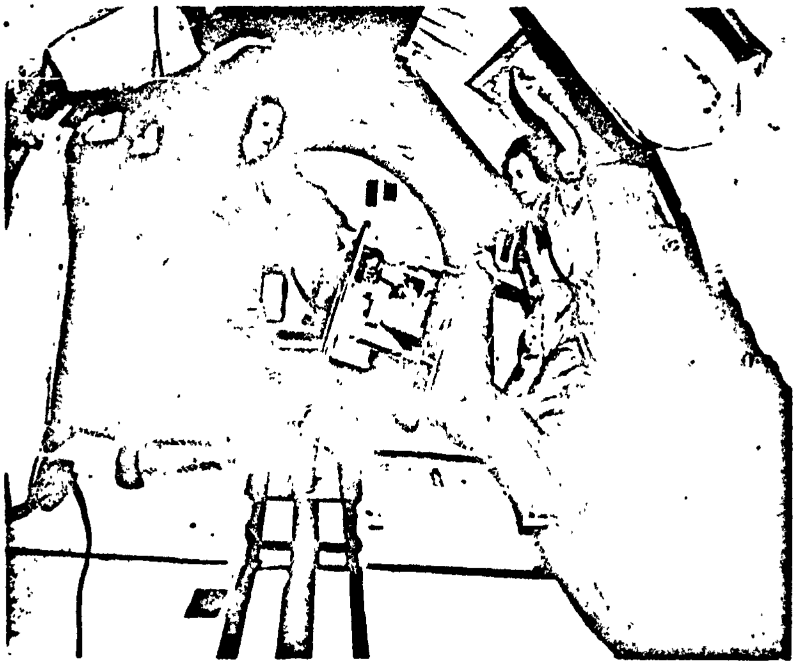


Figure 25. Technicians Releasing Test Capsule as It Becomes Weightless During a Zero "G" Parabolic Trajectory

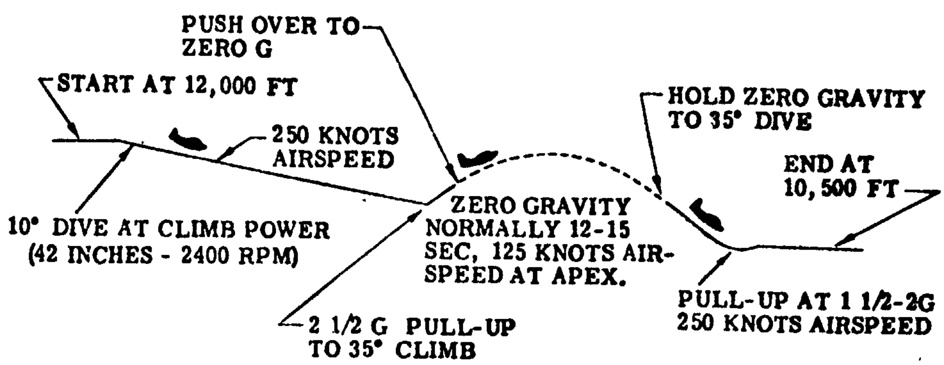


Figure 26. Parabolic Flight Path to Produce Zero "G" — C-131B Zero Gravity Facility

TABLE IV
Summation of Film Data

Run No.	Condition	Volume of Lubricant (cc)	Rig Speed (rpm)	View	Remarks
1-14	one "g"	15	-	-	Trying different lighting system, film types, frame speeds, etc
15	one "g"	20	2500	front	Lubrication appears adequate
16	one "g"	20	2000	end	Lubrication appears adequate
17	one "g"	20	2500	end	Same as Run 16 except action is a little more violent
18	one "g"	20	2000	front	Lubrication appears adequate
19	zero "g"	20	1500	front	No globules at first, then globules floating upwards. [illegible] drop of oil on plexi-glass. Sump nearly dry for last half of run
20	zero "g"	20	2000	front	More oil action than Run 19, between crankcase and top connecting rod pins. Plenty of oil on crankpins adhering and following. Better lubrication than Run 19. Large globule of oil adhering to lower left crankpin and rod
21	zero "g"	20	2700	front	About same as Run 20. Severe bump toward end of film shows oil being jarred mostly forward toward camera—i.e., oil had collected in the unswept area during zero "g" and then is jolted out and back into circulation
22	Transition to zero "g"	20	2000	front	Note angle of oil level in sump as rig is held at an angle before being released. Zero "g" follows. Oil moves forward toward camera near end of run
23	zero "g"	20	2700	end	Notice how oil collects in corners. Some action on gear teeth, but teeth appear to be carrying very little oil. Mechanism appears to be quite dry
24	zero "g"	35	2000	front	All unswept areas filled with oil. Some action of remaining oil. Note [illegible] out of way or scooping action of right hand flyweight on oil filled in flyweight cavity. Capsule hits cabin wall near end of run and [illegible] oil across face of flyweight cavity
25	zero "g"	35	2700	front	About same as Run 24. Some evidence of capsule jostling by fillet of oil in right hand flyweight cavity moving to the right
26	zero "g"	35	2000	end	Very first part of film shows evidence of some gravity. Transition to zero "g". Fairly stable operation and then rig hits cabin wall
27	zero "g"	35	2700	end	Not much oil visible in gear cavity. Some action visible in counterweight area

Discussion of Test Results

The test data verified the lubricant behavior to be as predicted—namely, there must be a sufficient volume of lubricant in the crankcase to fill all of the unswept volume within the crankcase and have a finite amount left to lubricate the moving mechanism.

It had been expected that correlation could be shown between splash patterns produced under one "g" conditions and those produced under zero "g" conditions. However, in reviewing the film data, it is difficult to establish a definite correlation—i.e., putting a certain amount of oil over and above the unswept volume of the crankcase in the rig and running may give indications of adequate lubrication under a one "g" condition, but may not be satisfactory for zero "g" running. For example, the 25-cc volume under a one "g" condition was more than enough to lubricate, and 35 cc under a one "g" condition was definitely too much; however, films taken during zero "g" operating conditions showed that 35 cc gave the best results.

Thus, it can be stated that the amount of lubricant necessary to lubricate a mechanism in a one "g" environment may not necessarily be sufficient to lubricate the mechanism under conditions of zero "g."

The film data clearly show the phenomena taking place within the crankcase. The lubricant collects in all of the unswept regions. When the entire unswept volume is full, and provided that there is more oil than "hiding space," the remaining oil is acted on or splashed about by the mechanism.

This was especially evident in several of the runs where the test capsule was jolted by striking the airplane cabin wall and a sudden acceleration or jolt to the lubricant resulted. The oil could be observed moving from its hiding places in the unswept areas and being struck by the moving mechanism.

Secondly, it can be suggested that for stabilized operation in zero "g" environment, besides the amount of lubricant needed to fill the unswept volume, sufficient lubricant must be present to allow film thicknesses on the moving parts to build up to the breakdown point, plus a small amount that will always be in droplet or globule form.

The films show this tendency of the oil to coat or form a thick layer on the moving parts when operating under zero "g" conditions, even though the parts are cycling at 42 cycles per second.

Consider a given droplet of oil traveling in a random direction until it strikes a moving object. Instead of breaking into several smaller droplets that go flying off in reactive directions, the droplet may simply be absorbed into the already existing oil film on the part. Droplets will continue to be captured by existing oil films until the thickness interferes with an adjacent part or the acceleration forces acting on the mass of the lubricant film are sufficient to overcome the surface tension and adherence characteristics of the lubricating media. As the lubricant breaks away, a droplet is formed and the cycle is complete. The droplet may be either absorbed or splashed during the next cycle.

In conclusion, it can be said that at best, it will be difficult to predict zero "g" lubricating characteristics from one "g" ground running tests. Therefore, it is suggested that space power lubrication systems, to be reliably operated for extended periods in zero "g" condition, should be tested in "weightless flight" aircraft prior to launching.

THERMODYNAMIC AND FLUID DYNAMIC COMPONENT TESTS

Single Heater Tube Pressure Drop Test

Continuation of this test to determine the effects of sharp corner conditions at the junction of the heater drilled passageway and the cylinder wall disclosed the previous results for sharp corners (reported in the last quarterly progress report) to be in error. Results reported in the last quarterly progress report were obtained with the access tube to the transducer filled with oil to reduce the dead space. The movement of this oil in the tube caused a pressure drop upstream from the transducer, and a change in Δ P across the tube was not observed with a change in piston speed as was previously reported. Accordingly, the oil level was reduced to the transducer dead space and the complete test was rerun.

Tests were conducted with zero tube end clearances at piston speeds of 1000, 2000, and 3000 rpm and with tube end clearances of 0.010, 0.020, 0.030, 0.040, 0.100, and 0.200 inch at 3000 rpm for both sharp corner and rounded edge conditions at the junction of the heater drilled passageway and the cylinder wall.

The Δ P in all cases increased with piston speed, and at 3000 rpm the maximum Δ P with sharp corners was 11.0 psi while the maximum Δ P with rounded edges was only 9.1 psi.

At 3000 rpm the Δ P remained constant for all cases as the tube end clearance was increased from zero to 0.200 inch.

Hence, it may be concluded that the Δ P across the heater tube increases with piston speed, that rounded edges at the junction of the heater drilled passageway and the cylinder wall decreased the Δ P across the heater tube by approximately 2.0 psi at 3000 rpm, and that an increase in the tube end clearance from zero to 0.200 inch (much greater than manufacturing tolerance) does not effect the Δ P across the heater tube.

Based on the results of these tests, the entrances to all heater and cooler tube passages of the PD-46 engine were rounded.

Unit Heat Exchanger Test

The unit heat exchanger test, discussed in the first quarterly progress report, has been completed with modified instrumentation.

During shakedown running of the test apparatus, it became apparent that the opening pressure of the check valves was too high for the pressure fluctuation obtained. Transducers with plus or minus capabilities were substituted for the check valves, and pressure signals were obtained with an oscilloscope. It soon became apparent that the pressure transducers used to obtain instantaneous pressure measurements were restricted at the connections. This was evidenced by the fact that the amplitude of the pressure variation decreased with increasing speed instead of increasing. An attempt was made to rectify this problem, but due to the physical shape of the test apparatus, the transducer inlet tube diameters could not be increased and the dead space decreased significantly to eliminate it entirely. Consequently, the results obtained were purely qualitative; but, as the test progressed, no significant changes were noted, and it was not necessary to pursue the restriction problem further.

Pressure differentials were obtained in the cylinder, at the bottom of the cooler, in the space between the cooler and the regenerator, and in the space between the regenerator and the heater tubes for all test runs. Data were taken at 1000, 2000, and 3000 rpm with the regenerator (1) against the inner shoulder on top of the cooler, (2) against the face in which the heater tubes are brazed, (3) against the top face of the cooler, and (4) midway between the cooler and the heater tube entrances, as shown in Figure 27. At the conclusion of the foregoing part of the test, the entrances and exits

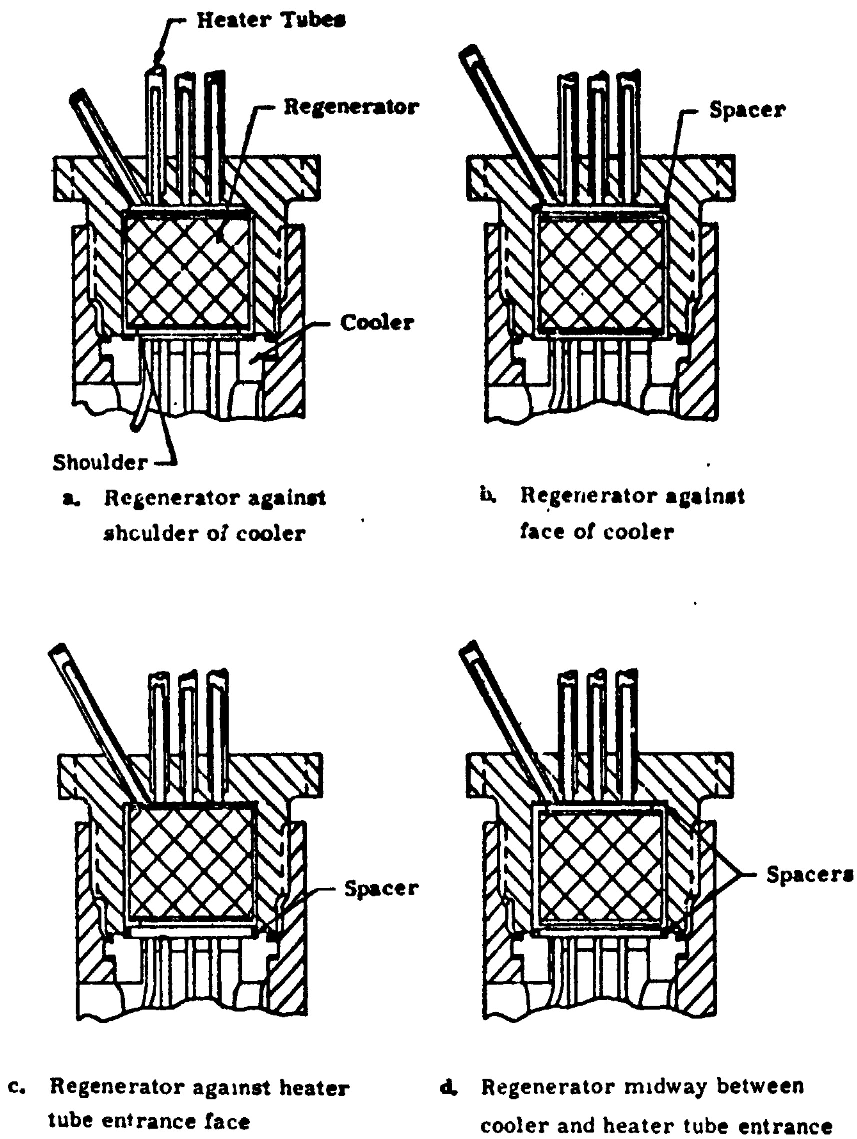


Figure 27. Locations of Regenerator for Unit Heat Exchanger Test

of the heater tubes and the tubes in the cooler were rounded off and the pressure differentials were again obtained with the regenerator against the face containing the heater tube entrances.

TABLE V

Pressure Differentials in Unit Heat Exchanger for Fluctuating Flow

Location of Regenerator	CPM	ΔP at Top of Cylinder (psi)	ΔP at Bottom of Cooler (psi)	ΔP Between Cooler and Regenerator (psi)	ΔP at Heater Tube Entrance (psi)
Regenerator against shoulder of cooler	1000	2.2	2.4	2.0	0.2
	2000	1.3	2.2	1.8	0.2
	3000	0.7	1.4	1.0	0.2
Regenerator against heater tube entrance face	1000	2.2	2.3	2.0	0.2
	2000	1.2	2.0	1.8	0.2
	3000	0.7	1.4	1.1	0.2
Regenerator against top face of cooler	1000	2.3	2.3	2.0	0.2
	2000	1.3	2.0	1.6	0.2
	3000	0.6	1.3	1.0	0.2
Regenerator midway between cooler and heater tube entrance face	1000	2.3	2.3	2.0	0.2
	2000	1.5	2.0	1.8	0.2
	3000	0.8	1.4	1.0	0.2
Regenerator against heater tube entrance face with rounded tube entrances and exits	1000	2.2	2.4	2.0	0.2
	2000	1.6	2.0	1.7	0.2
	3000	0.7	1.4	1.0	0.2

The pressure differentials for all of the conditions checked are shown in Table V. These data indicate that the pressure differential, at any particular location or speed, did not change appreciably regardless of the regenerator location or the condition of the tube entrance and discharge.

In order to verify the fact that there were no appreciable changes in pressure differentials, Bourdon gauges were connected to the pressure taps and steady flow readings were obtained for a range of air flow from 0.26 lb/min to 0.57 lb/minute.

The results obtained for the regenerator positioned against the cooler face, for the regenerator midway between the cooler and the face in which the heater tubes are brazed, and for the regenerator against the heater tube entrance face are shown in Table VI. Although the pressure differentials are slightly higher when the regenerator is displaced from the middle position, the penalty for doing so is not large enough to be of concern.

TABLE VI

Pressure Differentials Across Heat Exchanger Components for Steady Flow

Location of Regenerator	*Orifice ΔP (In. Kerosene)	W (lb/min)	Inlet Pressure (psi)	Total** Sidearm ΔP (In. Hg)	ΔP Across Regen-erator (In. Hg)	ΔP Across Heater (In. Hg)	ΔP Across Cooler (In. Hg)
Regenerator against face of cooler	80	0.578	34.5	65.0	19.9	9.80	32.25
	70	0.516	30.0	58.4	17.95	7.95	28.5
	60	0.457	26.0	48.7	16.10	6.40	24.14
	50	0.397	22.0	40.8	13.85	4.95	20.70
	40	0.331	17.0	32.0	11.60	3.40	16.07
	30	0.266	13.0	24.5	9.20	2.40	11.83
Regenerator midway between cooler and heater tube entrance face	80	0.564	32.0	60.4	21.0	9.40	27.65
	70	0.504	28.0	53.1	19.0	7.80	24.93
	60	0.444	24.0	45.0	16.7	6.10	19.98
	50	0.3851	20.0	37.8	14.4	4.70	17.05
	40	0.3251	16.0	30.0	12.35	3.40	12.75
	30	0.2641	12.5	23.0	9.8	2.40	11.23
Regenerator against heater tube entrance face	80	0.564	32.5	61.0	19.8	11.40	27.57
	70	0.504	28.0	53.0	17.8	9.40	23.10
	60	0.444	24.0	45.8	16.0	7.60	20.40
	50	0.385	20.0	37.8	13.75	5.75	16.34
	40	0.326	16.5	30.2	11.7	4.30	13.03
	30	0.264	12.5	23	9.5	2.90	10.00

*Orifice Size = 0.200 for all three conditions

**Includes pressure drop across connecting duct between cylinder and bottom of cooler

It may be concluded for the Unit Heat Exchanger Test that the position of the regenerator does not change the pressure differentials appreciably, regardless of whether the flow is fluctuating or constant and that the effect of entrance and exit losses due to sharp corner conditions at tube openings was not apparent.

V. MODEL PD-67 STIRLING ENGINE DESIGN

During the second quarter reporting period, design work has continued on the Model PD-67 Stirling cycle engine. In this period, design investigations have been conducted on two alternate configurations of the engine described in the first quarter report. The designs explored are:

1. Model PD-67A Engine
   This configuration consists of adapting the Model PD-67 cylinder head assembly to the Model PD-46 drive mechanism and crankcase.

2. Model PD-67B Engine
   This configuration consists of the Model PD-67 cylinder head and drive mechanism adapted to the Model PD-46 crankcase, modified for a zero gravity lubrication system.

The purpose of these investigations was to determine the advantages which could be achieved in over-all program time and cost while maintaining essentially the same project end objectives. This study showed that there may be some merit in the separate test evaluation of the engine major components. At the present time the relative merits of this approach and those of testing all major components simultaneously are being weighed. The final decision will be based on the effect of each approach on program timing and costs.

The detail design work conducted in the second quarter on the three basic engine subassemblies (cylinder head, drive mechanism, and crankcase) is described in the following paragraphs.

CYLINDER HEAD ASSEMBLY

The design layout and dimensional stack drawing for the Model PD-67 cylinder head assembly has been completed and the detail drawings for the various components are currently being prepared. Checking of the drawings is also in process and is partially completed.

The final design of the cylinder head, as shown in Figure 28, is somewhat different from the design described in the first quarter report. The changes were made in order that the latest Model PD-46 engine test information could be integrated into the flightweight design. With the new cylinder head configuration, it is anticipated that an increase in engine output and efficiency can be achieved.

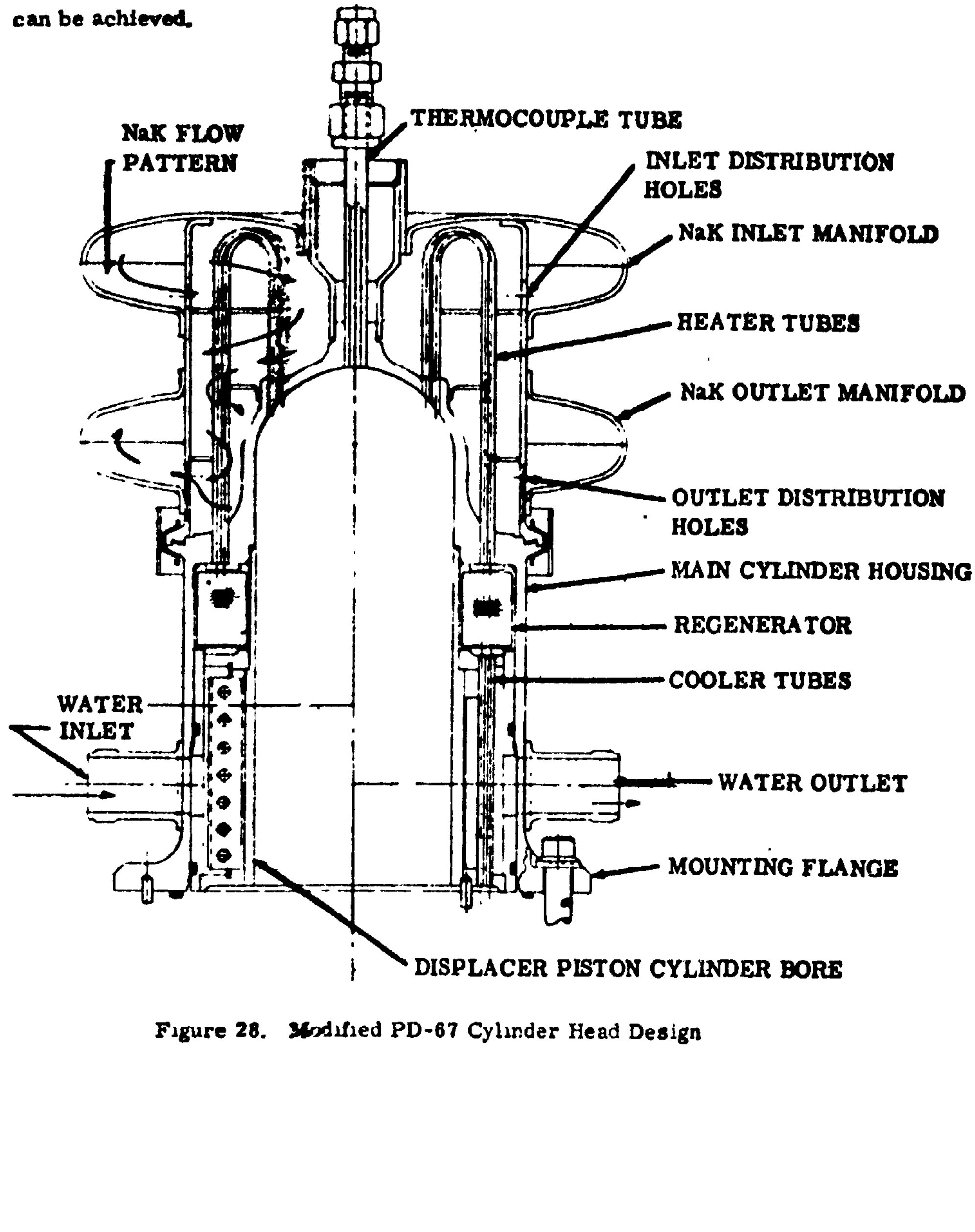


Figure 28. Modified PD-67 Cylinder Head Design

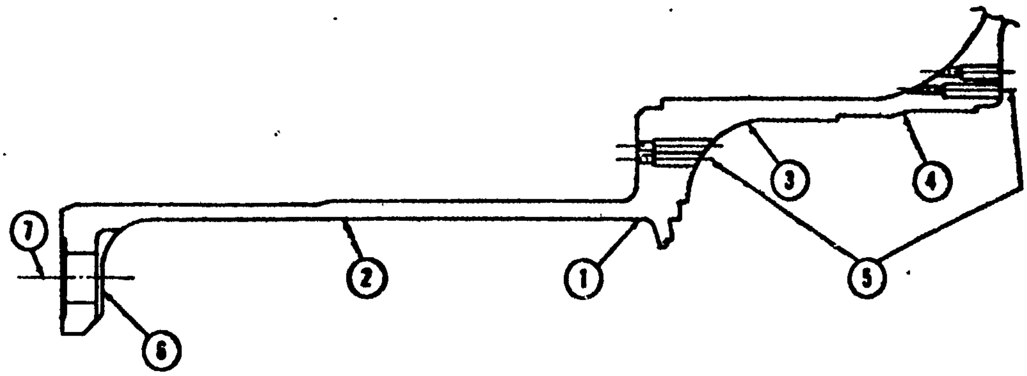
The new design incorporates 76 heater tubes arranged in two circumferential rows. The previous Model PD-67 design had 96 tubes arranged in three rows. The tube inside diameter has also been reduced from 0.072 to 0.060 inch and the length increased from 4.1 to 6 inches. By this change, a higher Reynolds number and greater tube l/d are achieved which should improve the heat transfer coefficient. The new design more nearly matches the tube proportions used in similar Philips Stirling engine designs. The NaK manifold has also been modified to accept the new heater tube configuration. An inlet manifold, identical with the outlet manifold, has been incorporated.

This revision simplifies the baffling arrangement and the connection for the expansion space thermocouple. NaK flow distribution is maintained by utilizing 12 small holes located in the partition plate adjacent to the outer collector chambers. The average NaK velocity in the tubes region is approximately 3 ft/sec and the pressure drop is estimated to be 3 psi.

The main cylinder dome height has been reduced. Analysis of the heat conduction path and thermal stresses have indicated this change to be beneficial from a weight standpoint. The mounting flange has also been modified to permit better piloting to the crankcase.

The stress analysis of this major engine subassembly has been completed. The location, magnitude of stress, type of loading, and estimated factor of safety for the cylinder head housing are shown in Figure 29. From this chart it may be seen that the numerical value of the factor of safety in certain locations is only slightly above unity. For example, at Location (1) the estimated value is 1.13. This value would indicate a somewhat marginal condition except for the fact that the allowable stress selected was based on the fatigue limit of the material rather than the yield point.

The cylinder head design specifications, which include the materials selected and the estimated weight, are listed in Table VII.



ITEM	CONDITION	STRESS (PSI)	LOADING	FACTOR OF SAFETY
1 WALL	MEAN PRESSURE WITH THERMAL EFFECT	34,000	FATIGUE	1.13
2 WALL	MEAN PRESSURE	24,000	FATIGUE	1.54
3 WALL	MEAN PRESSURE	32,000	FATIGUE	1.22
4 DOME WALL	MEAN PRESSURE	15,000	FATIGUE	2.25
5 TUBES	MAX. PRESSURE	500	SHEAR	37.6
6 FLANGE	120% PRE-LOAD	33,000	BENDING	1.61
7 BOLTS	120% PRE-LOAD	64,000	TENSION	1.57

Figure 29. Preliminary Stress Summary of Model PD-67 Engine Cylinder Head

TABLE VII

Model PD-67 Cylinder Head Design Specifications

1. General	
Design Type	Liquid Metal Heated
Heating Fluid	NaK (eutectic)
Arrangement	Annular Regenerator
Coolant Fluid	Water
Cylinder Bore Diameter	2.375 in.
Height - Over-all	9.030 in.
Width - Over-all	8.060 in.
Weight - Estimated	13.200 lb
Dead Space - Estimated	1.978 cu in.
Working Fluid	Helium (dry)
2. Heater Section	
Heater Tube	
Number	76
OD	0.090 in.
ID	0.060 in.
Length (Mean)	6.0 in.
Metal Temp (Mean)	1220°F
Pressure (Working)	
Max	1987 psi
Min	1101 psi
Helium Δ P (Mean)	6.35 psi
Helium Flow Velocity (Mean)	205 ft/sec
3. Regenerator Section	
Height - Cartridge	0.80 in.
Matrix - Screen	
Number (Estimated)	250
Wire Diameter	0.0016 in.
Mesh	250
Filler Factor	0.31
Area - Cross-sectional	6.24 sq. in.
Helium Flow Velocity (Mean)	15.3 ft/sec
Helium Δ P (Mean)	2.86 psi
Cartridge Plate Temperature	
Top	1215°F
Bottom	1730°F

TABLE VII (cont)

4. Cooler Section

Cooler Tube	
Number	152
OD	0.062 in.
ID	0.040 in.
Length	2.6 in.
Metal Temperature (Mean)	168°F
Pressure (Working)	
Max	1987 psi
Min	1101 psi
Helium Δ P (Mean)	3.83 psi
Helium Flow Velocity (Mean)	117.0 ft/sec

5. NaK System

Flow	25 gpm
Pressure - Inlet	25 psi
Pressure Drop	3 psi
Temperature	
Inlet	1256°F
Outlet	1221°F
Flow Baffles (No.)	3
Velocity (NaK)	
Inlet	13.32 ft/sec
Around Tubes (Mean)	3.41 ft/sec
Outlet	13.32 ft/sec

6. Coolant System

Flow	5 gpm
Pressure - Inlet	75 psi
Pressure Drop	2 psi
Temperature	
Inlet	150°F
Outlet	163°F
Flow Baffles (No.)	2
Velocity (Coolant)	
Inlet	5.22 ft/sec
Around Tubes (Mean)	2.38 ft/sec
Outlet	5.22 ft/sec

TABLE VII (cont)

7. Materials

NaK Manifold	AMS 5521 (310)
Cylinder Housing	AMS 5754 (Hastelloy X)
Heater Tubes	AMS 5536 (Hastelloy X)
Regenerator Shell	AMS 5521 (310)
Regenerator Matrix	AMS 5697 (304)
Cooler Housing	AMS 5521 (310)
Cooler Tubes	AMS 5572 (310)

DRIVE MECHANISM

The design layout of the Model PD-67B drive mechanism was completed during the second quarter. The dimensional stack drawing is currently in process. Detail drawings of the various components will not be initiated until ASD approval is obtained.

All recent design effort on the crank linkage has been directed towards the Model PD-67B configuration. This design approach required that changes be made to the drive mechanism described in the first quarter report. The modified crank and piston assembly is shown in Figure 30. Essentially, both piston shafts have been lengthened in order that this assembly could be incorporated in the Model PD-46 crankcase. The two pistons were also sized for the Model PD-67B configuration. All other components are adaptable to either the Model PD-67 or PD-67B engine.

The design refinements (refer to Figure 30) which have been made to the basic assembly are as follows:

1. Displacer Piston
   A white metal seal band is utilized instead of a Teflon ring for the seal. Based on test experience obtained from the Model PD-46, this type of seal has indicated adequate seal performance in regard to gas leakage, seal friction, and life.

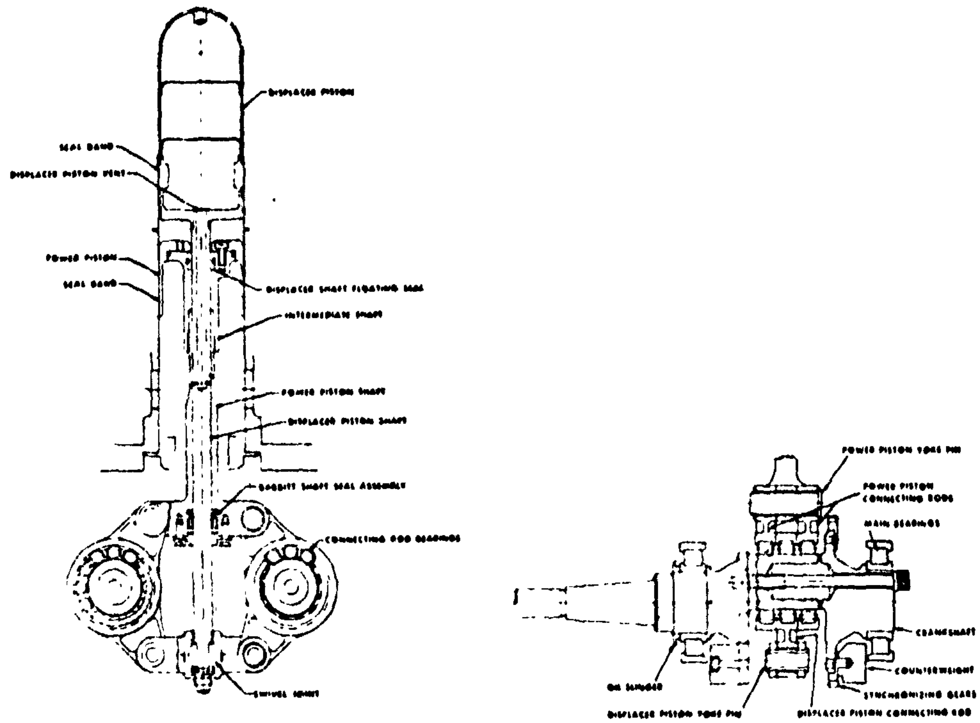


Figure 30. Modified Crank and Piston Assembly

One additional change has been incorporated in the displacer piston. The capillary tube, previously vented to the working zone, is now vented to the buffer zone via a drilled hole in the displacer piston shaft. In this design, the piston is completely sealed from the working zone which permits optimum output to be achieved.

2. Power Piston

The power piston in the latest design utilizes a white metal seal instead of three Teflon rings. This change was made as a result of satisfactory operation obtained on the Model PD-46 engine equipped with this configuration. Also, the piston skirt has been lengthened and the skirt wall thickness decreased. Improved sealing characteristics should be obtained by these modifications.

3. Swivel Joint at Displacer Piston Yoke

As shown in Figure 30, a swivel joint has been incorporated in the displacer piston yoke. In addition, the displacer piston shaft is bushed at the lower end of the power piston shaft yoke. By this arrangement, improved control of the displacer shaft motion can be achieved. The high bending stresses at the yoke joint were also eliminated.

4. Crankshaft

The lubrication holes incorporated in the original model PD-67 crankshaft have been deleted. A new lubrication system has been used in the engine and these holes are no longer required. The revised lube system is described in a later paragraph in this section.

The final configuration has been checked for balance and found to be satisfactory. Preliminary stress analyses have been conducted on the crankshaft curvic coupling and journals. In all cases the design has been found to be conservatively stressed. The maximum crankshaft deflection is less than 0.0008 inch under loaded conditions.

Both the connecting rod and the main bearings have a calculated B-10 life of approximately 100,000 hours when using premium steel. This life is considerably higher than that required for one year's reliable operation.

The preliminary design specifications for the Model PD-67B drive mechanism are listed in Table VIII.

TABLE VIII

Preliminary Design Specifications for the Model PD-67B Drive Mechanism

1. Crankshaft

Type	2 piece
Connecting Rod Journal Diameter	1.4756 in.
Main Journal Diameter	1.5749 in.
No. of Counterweights	2
Crank Radius	0.516 in.
Curvic Coupling	
OD	1.440
Thickness	0.180 in.
No. of Teeth	24
Type	Concave-convex
Synchronizing Gear	
Diametral Pitch	16
Pitch Diameter	4.375 in.
No. of Teeth	70
Face Width	0.245 in.
Main Bearing	
Type	[illegible]
Width	[illegible] in.
ID	[illegible]
OD	[illegible]

2. Connecting Rod

No. of Rods	4 Power 2 Displacer
Type	Solid
Rod Length (Center-to-Center Distance)	2.062 in.

TABLE VIII (cont)

Bearings	
Type	Roller
No. of Rollers	12
Roller Width	9 mm
Roller Diameter	9 mm
Bushings	
Type	Sleeve
ID	0.5625
Length	0.580 in.

3. Piston Shafts

Power Shaft	
Type	Hollow
OD	0.8401 in.
ID	0.530 in.
Length	7.029 in.
Seal Type	Spring-loaded Ring
Displacer Shaft	
Type	Solid
OD	0.5200 in.
Seal Type	Spring-loaded Ring

4. Power Piston

OD	2.375 in.
Length	2.000 in.
Seal Type	White Metal

5. Displacer Piston

Type	Pressurized
OD	2.375 in.
Length	7.560 in.
Seal Type	White Metal

6. Material

Crankshaft	AMS 6260 (9310) & AMS 6415 (4340)
Connecting Rods	AMS 6260 (9310)
Power Piston Shaft	AM 6415 (4340)
Displacer Piston Shaft	AMS 6415 (4340)
Displacer Piston Shaft Yoke	AMS 6415 (4340)
Power Piston	AMS 6415 (4340)
Displacer Piston	AMS 5521 (310)

Crankcase

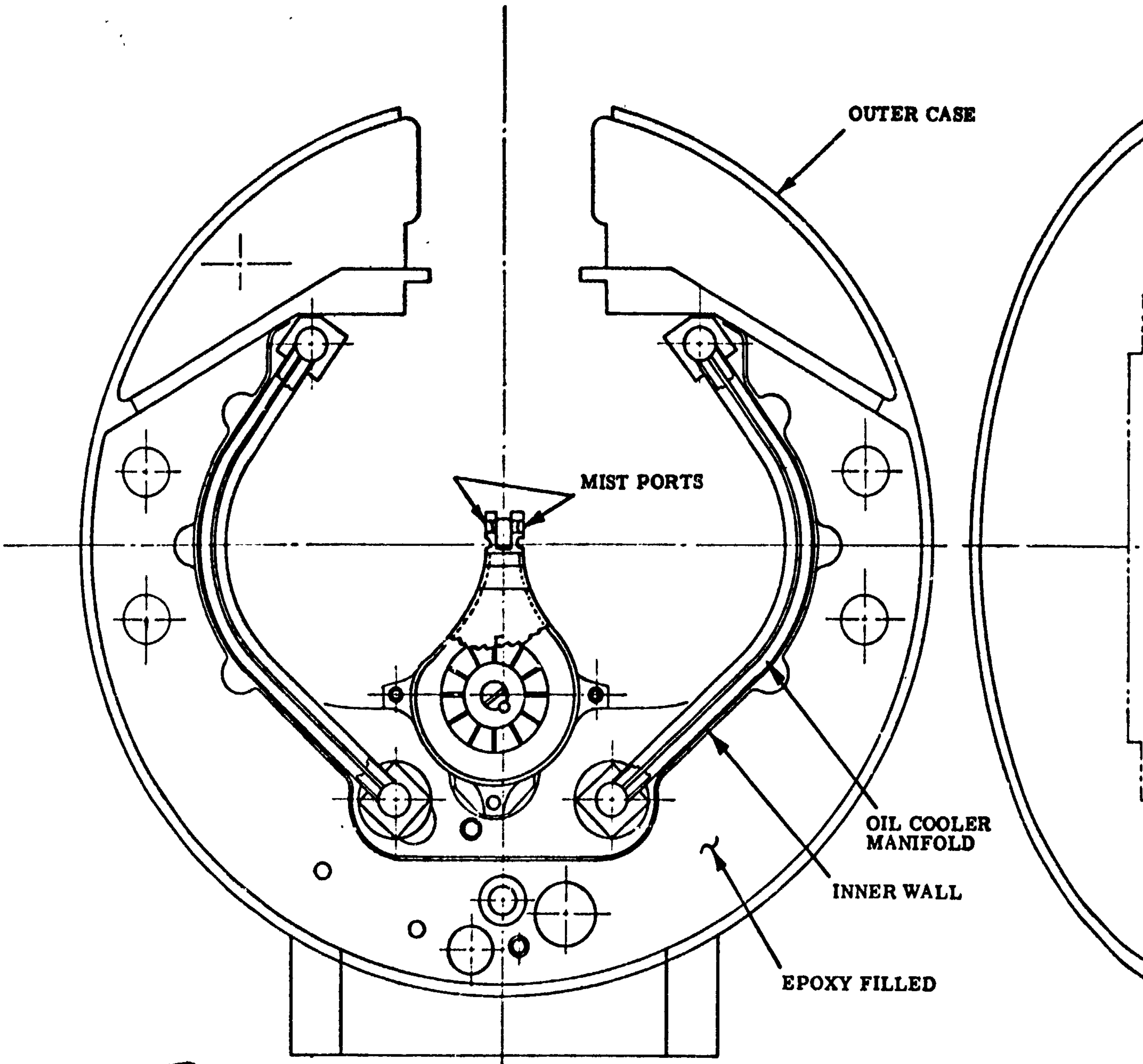
The Model PD-67B crankcase design is currently in process. During the second quarter reporting period all effort has been directed toward the design modifications required of the Model PD-46 crankcase to accept the flightweight drive mechanism and the zero gravity lubrication system. The partially completed crankcase layout is shown in Figure 31.

The existing model PD-46 crankcase has been revised to accept an inner wall in the center section. The cavity between the inner and outer walls is filled with epoxy to withstand the crankcase operating pressure. Mounted within the inner wall are two oil cooler manifolds similar to the ones described in the first quarter report.

The main bearing plates have been redesigned to accept the rolling contact main bearings of the Model PD-67B drive mechanism. Although it is not shown, the upper shaft seal wall also will be redesigned commensurate with the Model PD-67B shaft requirements.

A new type of zero gravity lubrication has been integrated into this case design. The system includes a small centrifugal fan, collector housing, and two jets which direct the discharged oil mist into the connecting rod assemblies. The fan is geared to the crankshaft timing gear and operates at a speed of 7500 rpm. The fan shaft is supported by two ball bearings which are lubricated by oil mist. The fan housing is bolted to the crankcase bulkhead.

The system operates in the following manner. The oil throw-off from the drive mechanism flows into the eye of the centrifugal impeller. The oil mist is pressurized to approximately ten inches of water and collected in the housing. The mist is then forced out through two jets located in the upper part of the housing. Two streams of oil mist are thus directed on the


OUTER CASE
MIST PORTS
OIL COOLER MANIFOLD
INNER WALL
EPOXY FILLED

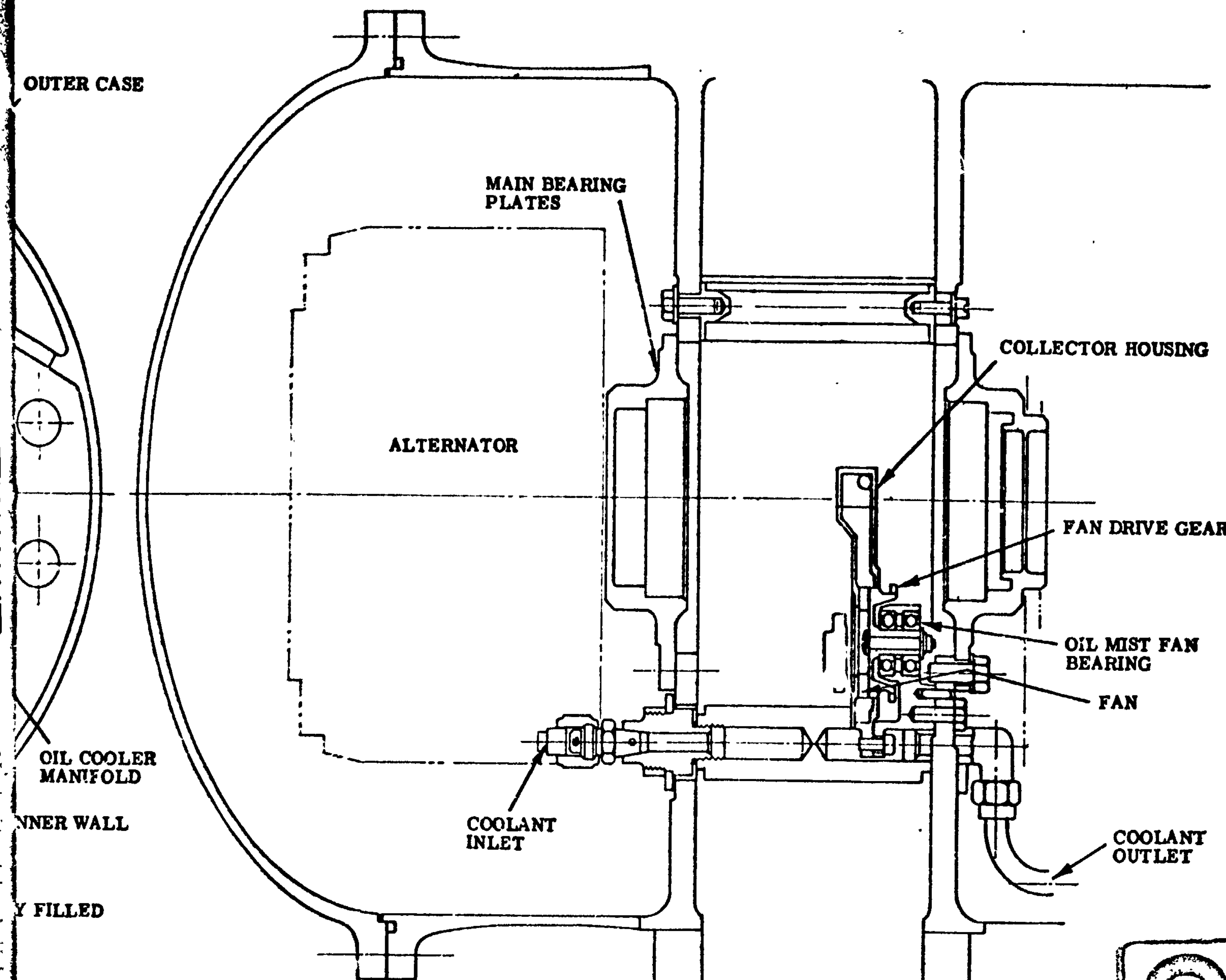


Figure 31. Partially Completed Crankcase Layout

connecting rod parting line. Oil enters the connecting rod bearings through slots milled in the connecting rod face. After careful design analysis, this system, which is a more positive approach to the lubrication problem while operating in a gravity-free environment, was selected over that previously described.

VI. PROGRAM SCHEDULE

MILESTONE CHARTS

The progress to date of the various tasks of this program is shown in Figures 32 through 35. The broken bars on these charts represent the original program schedules, the cross-hatched bars represent revisions to these schedules, and the solid bars indicate actual work accomplished.

The original program schedules (broken bars) for Tasks 1, 2, 3, 5, and 7 are consistent with the milestone charts contained in the Technical Proposal, EDR 2025, dated 9 March 1961. The original program schedule for Task No. 4 does not agree with the schedule contained in the original proposal. This difference is the result of the negotiations between Allison and ASD which took place after submittal of the proposal, but before execution of the present contract. The original program schedule for Task No. 5, "Advanced Flight Type Engine Design," is consistent with the Project Control Chart included in the Technical Proposal, EDR 2025. A milestone chart for this task was not included in the proposal.

All work scheduled for Tasks 1, 2, 3, and 7 has now been completed. Task 4 was originally scheduled for completion by the end of the second quarter; however, additional testing of the PD-46 engine will be required. Although performance improvements were realized in the second quarter, additional testing and analytical studies will be required to ensure the desired performance capabilities of the flight-type design. Tests have been scheduled through the first half of November 1961, and additional tests will be scheduled at a later date if required.

Several changes will be noted in the milestone chart for Task No. 6. In the first quarterly report, it was predicted that all design and drafting of the

cylinder head assembly would be completed approximately six weeks ahead of the original schedule. This was not accomplished because of the recent indication of the possibility of unfavorable flow conditions in the heater tubes. The heater section has been modified and, as indicated in Figure 34, all drafting and checking will be completed about one-half week later than the original schedule date. The change in the schedule for the drive mechanism and crankcase is a result of the current analysis of the feasibility of the PD-67A and PD-67B configurations (PD-67 cylinder head and drive mechanism and PD-46 crankcase). The increased time estimated for detail drawings and checking is the result of a decision to assign fewer men to this phase of the program and does not involve an increase in the scope of work to be accomplished.

APPLICATION OF "PROGRAM EVALUATION PROCEDURE" (PEP)

A revised PEP chart is not included in this report because of the probable redirection of the future program. A revised chart will be prepared when the program revisions are finalized.

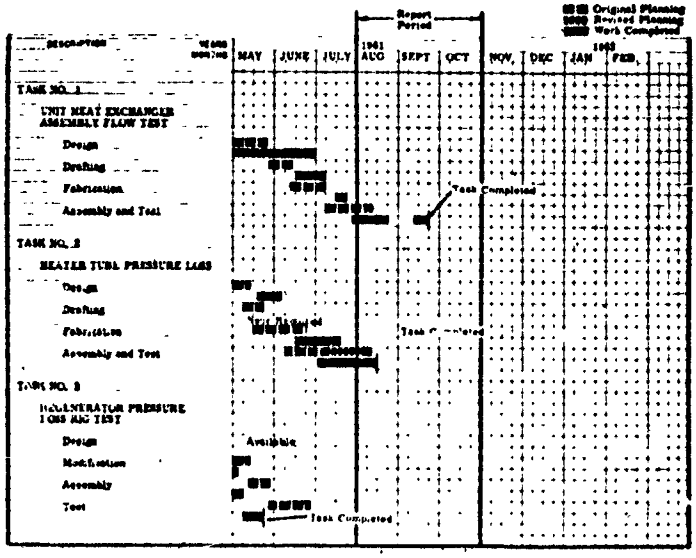


Figure 32. Milestone Chart — Tasks 1, 2, and 3

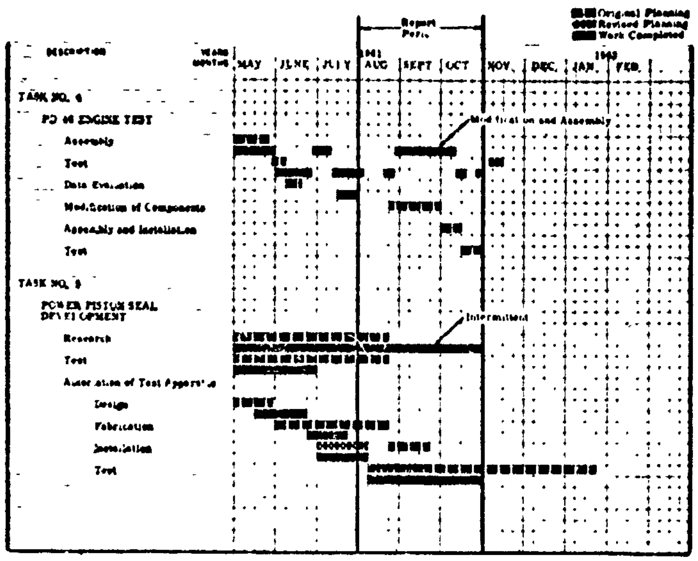


Figure 33. Milestone Chart — Tasks 4 and 5

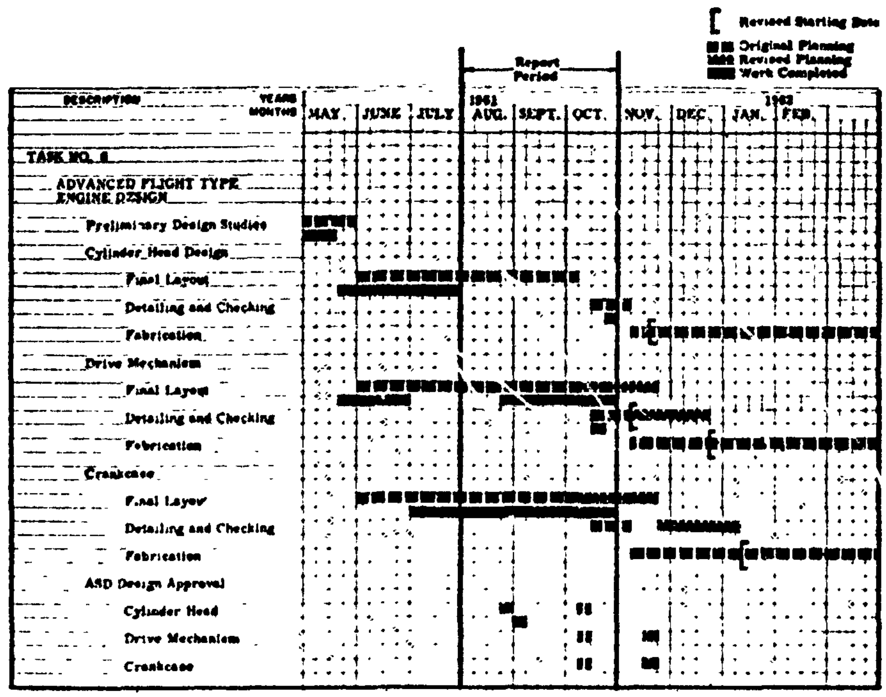


Figure 34. Milestone Chart — Task 6

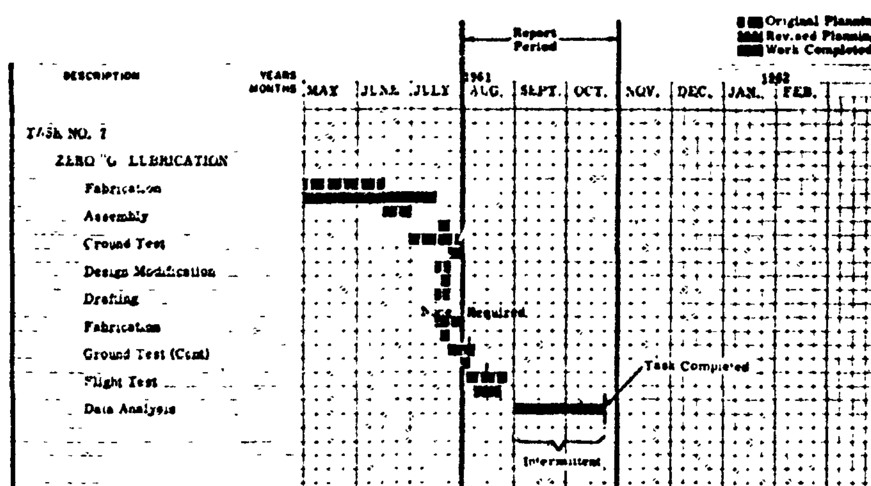


Figure 35. Milestone Chart — Task 7